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TRANSPORTATION RESEARCH COMMAND
FORT EUSTIS, VIRGINIA

TCREC TECHNICAL REPORT 62-41

THE APPLICATION OF MODULAR/SECTIONAL STRUCTURES TO GROUND EFFECT MACHINES

Task 9R99-01-005-06
Contract DA 44-177-TC-727

June 1962

prepared by:

BOOZ ALLEN APPLIED RESEARCH, INC.,
Bethesda, Maryland

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This report presents the results of an investigation conducted by Booz-Allen Applied Research, Incorporated, under Project 9R99-01-005-06, GEM Structures Study.

Since the anticipated employment of air cushion vehicles will of necessity be mission oriented, consideration should be given at the earliest possible moment to methods of transporting them to critical mobility areas where their use is indicated.

Size constraints of existing carriers, other than ocean-going vessels, are generally incompatible with the planform dimensions of practical air cushion logistical vehicles. The intention of this study was to analyze those size constraints and to investigate methods of sectionalizing, collapsing, or modularizing a typical air cushion vehicle structure to permit movement to and within the overseas theater of operations.

The study also indicates the effect of such treatment on the weight, configuration, performance, and efficiency of the vehicles considered.

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Task 9R99-01-005-06

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TCREC Technical Report 62-41

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for
U. S. ARMY TRANSPORTATION RESEARCH COMMAND
FORT EUSTIS, VIRGINIA

PREFACE

The study presented in this report was undertaken by the Aerospace Engineering Group, Booz·Allen Applied Research, Inc., 4815 Rugby Avenue, Bethesda 14, Maryland, and sponsored by the U. S. Army Transportation Research Command under Contract Number DA 44-177-TC-727, Job Order Number 130-6. This research program was carried out under the supervision of Mr. Peter G. Fielding. This study began in July 1961 and was completed in March 1962. Mr. W. E. Sickles of the U. S. Army Transportation Research Command was the Project Officer.

This contract directs the contractor, among other things, to determine the most suitable structural materials, configurations, and standards for GEMs in the amphibious regime of transportation for the U. S. Army. It directs particular attention to the problems of air and surface transportability.

The authors are indebted to the personnel of the GEM Group at USATRECOM for their guidance and cooperation and also to the Transportability Group at USATRECOM and to Mr. Carl Weiland, GEM designer formerly with Reynolds Metals, and Maj. Lester Robertson, USATRECOM Liaison Officer at ONR, for the helpful discussions held with them.

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SUMMARY

This report contains the results of an investigation of structures for a Ground Effect Machine (GEM) which would be useful in the Army's amphibious regime of transportation. Particular emphasis is placed on arriving at a structural concept which would produce a vehicle with maximized transportability. A vehicle for the Logistics-Over-The-Shore (LOTS) mission, the requirements for which are fairly well defined at this time, is selected so as to have a concrete example for design purposes.

As a prerequisite to vehicle engineering, the mission, the use sequence, and the characteristics which would make the vehicle useful and transportable are examined in detail. The loading doors of transport aircraft and trucking limitations are found to be the most restrictive. The mission for the vehicle indicates that low density cargo will predominate but that a variety of military equipage should be provided for and that a simple design with open cargo space is preferred.

Before the design studies proper, there is additional necessary material for guidance. The current state of the art is surveyed. It is found that aircraft-type structures are in general use in GEMs but that the proportion of vehicle gross weight devoted to structure varies widely among the different engineering organizations. An attempt is made to understand the variation through consideration of criteria on which the designs are founded. This produced results which are only moderately conclusive, probably due to the fact that secondary structure is present in the vehicles to various degrees, depending on how the lifting system is disposed, how much structure overhangs the cushion area, and like matters. Structural criteria are proposed following a consideration of all the operational, laboratory, test, and specification-type data that are pertinent. The most severe loads to be resisted are believed to be those associated with operating in the military amphibious environment rather than those of normal over-water operation. The miscellaneous environmental requirements are enumerated. Material properties appropriate to this kind of application are studied and a selection made -- aluminum alloy 6061-T6 -- primarily on the basis of its corrosion resistant property which weighed heavily in the optimum of all those with high specific strength.

A preliminary design process is executed. Power requirements are determined as a function of gross weight, cushion pressure, and operating height. Conventional analyses are employed and the results are in accord with the state of the art, although installed power is toward the high end of the scale because of the high surface clearance necessary under certain adverse operating conditions and the high cushion pressure necessary to keep bulk down. The lifting system is highly simplified and contains minimum ducting.

In comparison, a modular approach to design and vehicle layout which anticipates modularization is found much superior to attempting the sectionalization of an existing design. However, after the effort to design for maximum transportability as a requirement is superposed on all others, it is found that the structure is of quite low density. This does not inhibit its transportation, but it does mean that transport vehicles lose efficiency when carrying this kind of cargo. An attempt to use the modules in alternate arrangements to provide a series of sizes of vehicles showed disappointing performance for the "off-design" combinations. Detailed stress analysis is performed to refine the weight figures which were only approximate in the first instance. Increases to the initial weights are nominal. Assembly and field problems are considered and no out of the ordinary difficulties can be foreseen. In a final weight review it is shown that means exist for decreasing weight, and for each weight saving the associated design changes are discussed.

Recommendations are included, the principal one being that the modular concept of vehicle design produces some definite advantages and should be employed in development programs for GEMs in the general category of the LOTS carrier.

CHAPTER I

TRANSPORTABILITY LIMITATIONS AND GENERAL MISSION REQUIREMENTS

1.1 ROLE OF VEHICLE

The study reported here is concerned primarily with a GEM vehicle whose requirements are relatively well defined at this time -- an amphibian designed primarily for logistic-over-the-shore (LOTS) operations and capable of transporting 15 tons of cargo over a range of 50 miles. Published reports of earlier GEM studies have indicated that this type of vehicle can reasonably be expected to make a real contribution to the mobility of military forces. That effort will not be duplicated in the present program; instead, the orientation in the present program is toward mobility of the LOTS carrier itself. Transportability of such equipment makes a contribution to strategic mobility -- the ability to move significant forces to any point in the world with minimum delay -- a requirement that calls for adequate air, land, and sea transport, and, above all, for highly transportable materiel.

The 15-ton carrier, while primarily an amphibian for LOTS operations, will also be capable of use on inland waterways and across country. It is expected to make possible military operations in remote and otherwise inaccessible areas. Under adverse conditions, the vehicle can be expected to sustain a speed of from 25 to 40 miles per hour; its optimum speed, as determined from technical considerations only, is much higher.

A comprehensive description of LOTS operations is provided by various current Army staff manuals (for example, Reference 1). From technical studies such as Reference 2, the developing trends can be inferred. All the data point to one major conclusion: supply movements over beaches can be expected to increase in importance as port vulnerability to nuclear weapons increases. Lighters or amphibious craft for unloading ships standing off-shore will therefore play a greater role than they would if intended solely for support of amphibious assault operations. Such craft must carry bulk cargo, palletized cargo, containerized cargo, vehicles and major equipment items, and

personnel. In short, these craft will be capable of handling the major resupply for an active overseas theater operations.

Naturally, some characteristics of LOTS operations are not precisely determinable in advance of their actual occurrence. The off-shore distance of the ships will depend on the size of the operation, the weather, enemy activity, etc. The inland distance to the unloading point will depend on the terrain, progress of operations, and availability of all types of transport. The total time required for the craft to perform one full cycle of operation will depend largely on the combined overwater and overland distances, but will also be affected by the unloading rates at the ship's cargo hatches and at the inland resupply point, by queuing operations, and by other sources of lost time. The contemplated distances range as high as 30 miles for off-shore distance and 10 miles for inland distance. If GEMs are selected for the role of LOTS carriers, it will probably be for the advantage of speed that they offer; and speed would be no major asset unless distances in the upper portion of the indicated range were contemplated. In this study, for planning purposes, a combined over water-overland distance of 25 miles, or a round trip distance of 50 miles, was arbitrarily assumed.

On the whole, the environment in which LOTS operations take place must be regarded as unfavorable. The major equipment item in the system -- the deep-craft, ocean-going vessel -- has evolved slowly over the years and is geared to cargo transfer at dockside or in sheltered harbors. Major changes in this characteristic are not expected within the next ten years. The exigencies of war entail operations in unfavorable weather (and the sea can multiply the effect of weather), at poor locations, and with incompletely trained personnel. Consequently, equipment designed for LOTS operations must be sufficiently rugged to permit successful performance of the mission under adverse environmental conditions. This subject is discussed more fully in a later section of this report.

1.2 MISSION DATA

In order to develop relatively firm and realistic working models for this study, it was first necessary to define the kinds of operations the vehicle is intended to perform. The primary mission must take first priority during development, but it is often possible to incorporate design features to give the vehicle flexibility and versatility. Hence,

a number of secondary or alternate missions were considered, as well as the primary mission.

A summarization of mission data for the 15-ton amphibious GEM designed for use as a LOTS carrier is presented below.

1.2.1 Primary Missions

- LOTS carrier normally arrives at locale of amphibious operation aboard ocean-going transports.
- Carrier is lowered to water fully assembled.
- Cargo is transferred to LOTS carrier by booms of transport.
- Handling lines and low propulsive power are used for steadying carrier during loadings.
- Fuel for hovering during loading is not available.
- Typical range without refueling is 50 miles, i.e., one round trip from ship to over shore and return.
- Low-density cargo down to 75 pounds per square foot is to be accommodated.
- High-density cargo up to 300 pounds per square foot is to be accommodated.
- Containerized cargo is to be sized: 6 feet 3 inches wide, 6 feet 10-1/2 inches high, 8 feet 6 inches long.
- Off-loading is to be accomplished -- generally by fork lift.
- A ramp will facilitate off-loading by fork lift or by manual effort.
- Typical passage might be in the range from 10 per cent to 60 per cent overland, the remainder over water, depending on the nature of the operation.

- Surf up to five feet in height must be negotiated; reduction of cargo to achieve operation at maximum wave height is to be expected.

- An alternate use is in lightering unit personnel and light unit equipment ashore.

- Maintenance operations are to be minimized in view of possible absence of 3rd and 4th echelon.

- Speeds in the range of 25 to 40 miles per hour are consistent with LOTS operations.

- Movements of LOTS carriers by air transport (Phase III airborne operations), either inter-theater or intra-theater, is desirable.

- Carrier shall be capable of sustaining and operating with a threefold overload.

1.2.2 Secondary Missions

- Types of operations:

  - Movement over inland waterways;

  - Movement from off-shore bases;

  - Penetration into land areas prohibitive to conventional equipment.

- Movement of tactical units on the carrier may be equal to or greater than movements of supplies.

- Maximum diversity of palletized loads and vehicles is to be accommodated.

- To expedite vehicle handling, ramps fore and aft are desirable.

- Personnel may also be carried more frequently.

- Provisions for fueling to increased ranges up to 200 miles shall be included.

- Higher echelon maintenance may be possible.

- Operation for extended periods at reduced surface clearances is likely.

- Benefits of operation in closer proximity to the ground in the form of either reduced fuel consumption, greater payload capacity, or greater range.

- Carriers may be frequently knocked down and transported overland by truck and/or train.

- Transport by air is more likely than in the case of LOTS operations.

1.3 GEM VEHICLE EMPLOYMENT SEQUENCE

There are other phases in the employment of the vehicles besides actual accomplishment of missions as defined in the foregoing sections. The entire sequence of uses must be considered to assure that any factors that might influence the design are not overlooked. The additional phases are as follows:

- Depot storage. Adequate preservation and protection from environmental extremes should be expected for long-term storage.

- Training use. Considerable numbers of vehicles must be assigned to new operator training; also, unit training will be necessary and may take place under deliberately rigorous conditions.

- Inactive periods. These periods often are a hazard to mechanical equipment.

- Maintenance operations. The design must be conducive to simplified maintenance.

- Deployment. Administrative moves of this size vehicle will most probably be made by rail and water; partial knockdown.

- Tactical moves. These invoke maximum transportability features of design, including rapid reassembly.

- Tactical employment. Mission summaries give the details.

1.4 ENGINEERING MILITARY EQUIPMENT FOR TRANSPORTABILITY

In accordance with DOD instructions, the military services have established a systematic program of transportability engineering to assure that items of military equipment are so designed and constructed that they may be efficiently handled by the available modes of transport. Policies governing the program do not rigidly specify requirements but recognize the possibility of conflict between optimized transportability of equipment and operational capabilities. Reference 3 contains general and detailed provisions; the main points are listed below.

- Transportability is a main consideration in assigning priorities to equipment characteristics.

- Designs should meet operational characteristics while adhering to fundamentals of transportability.

- Restriction to specific modes of transport is possible if this is consistent with end use.

- Disassembly or knockdown may be specified to solve transportability difficulties.

- Transportability engineering includes provisions for protection of fragile or dangerous articles.

- Adequate fittings for lifting, tie-down, and handling facilitate transportability.

- Designs should recognize gross weight limits and outside dimension limits.

- Highway weight and size limitations are established in the Federal Aid Highway Act (1956) and a majority of state laws:

Height:	150 inches, continental U.S.; 132 inches overseas areas.
Width:	96 inches.
Length:	35 feet, single vehicle; 50 feet, truck tractor and semitrailer.
Weight:	16,000 pounds per axle spaced less than 7-1/2 feet apart.
(on axles):	18,000 pounds per axle spaced more than 7-1/2 feet apart (overseas, 16,000 pounds is upper limit).
Weight: (gross)	36,000 pounds if front to rear axle distance is less than 10 feet.
	850 pounds allowed per foot of additional separation; 60,000 pounds is upper limit.
Size:	(for equipment loaded in van or stake-body truck) 6 feet 5 inches high, 7 feet wide, 22 inches long, 11,200 pounds weight.

- Railroad limits are based on outline diagrams which provide safe clearance for bridges, tunnels, station facilities, etc. AAR limits prevail in North America and Berne International in most of Europe, the latter being more restrictive (see Appendix A illustration). For AAR limits, 10 feet 11 inches is limiting height above car floor and 10 feet 8 inches is outside width. Corners of this rectangle above 9 feet 7 inches are not useable. Floor is 4 feet 2 inches above rail. For Berne limits, outside height is 9 feet 10 inches, outside width 10 feet 4 inches. Corners of this rectangle

above 6 feet 2 inches are not useable and car floor is again 4 feet 2 inches above rail. No weight limits are stated.

- Ocean-going vessels may be Mariner, Victory (illustrated in Appendix B), or Liberty class. Each of these has hatch openings of 22 feet x 30 feet or larger. Clearance under hatch girders varies for the different compartments and is a minimum of 7 feet 7 inches, 6 feet 7 inches, and 7 feet 10 inches for the three classes, respectively. Each vessel has numerous compartments with clearances between 10-11 feet. No weight limits are stated.

- Data on cargo airplanes consist mainly of dimensions for the cargo space and access doors and allowable weight for a normal 1,000 nautical mile (one-way) mission. Furthermore, it is cautioned that the full space of the cargo compartment is not available because access to the rear of the airplane must be allowed. Charts are included in Reference 3 which assist in the determining sizes and shapes passable through the doors. For illustration, the C-133 chart is reproduced as Appendix C. The basic size (approximate) and weight data are:

Airplane	Cargo Compartment	Door	Weight (normal)
C-119G	8H x 9W x 36L	8H x 9W	13,630 pounds
C-123B	8H x 9W x 36L	8H x 9W	11,785 pounds
C-124C	11H x 11W x 77L	17° ramp	47,600 pounds
C-130A	9H x 10W x 41L	9H x 10W	29,500 pounds
C-133A	12H x 11W x 97L	12H x 12W	95,000 pounds

(Note: All dimensions are feet.)

1.5 ADDITIONAL GUIDES TO TRANSPORTABILITY ACHIEVEMENT

Fitting a cargo to its carrier is only one aspect of transportability. A number of other matters are taken up in other sources. Reference 4 is particularly concerned with air transportability. Airborne operations are defined according to the following phases:

> Phase I (parachute and assault landing): Airborne divisions and their support. Parachute drop and/or assault landing are used. Materiel must be employed immediately; for certain selected exceptions, one hour is the outside limit. Assault-type aircraft are used.
>
> Phase II (initial air landing): Phase I follow-up and initial elements of the infantry divisions. Minimum criteria landing fields, in friendly hands, are used. Aircraft are therefore medium transport or assault type. Materiel must be ready for employment in one hour except for selected exceptions up to two hours.
>
> Phase III (heavy air landing): Follow-up forces for earlier phases. Heavy transport aircraft are used on prepared fields. Materiel should be ready for employment in six hours.

In addition to these airborne operations, there are also deployments by air which are administrative in nature. The landing facilities would be equal or superior to those encountered in Phase III operations and the six-hour time limit would prevail. No further requirements would be introduced on the materiel to be transported.

The required quantity of an item of materiel is pertinent information in planning for air transportability. Assume, for estimating purposes, a typical Field Army of 250,000 requiring 64 pounds per man per day of supply (per FM 101-10) or a total of 8,000 tons per day. Assume also that 80 tons could be moved per day by one LOTS carrier, this being a fairly conservative value; then 100 vehicles would be required. For a one-division operation, 10 vehicles is a reasonable estimate. See paragraph 7.5 for the airlift equipment.

1.6 THE CRITICAL TRANSPORTABILITY PARAMETERS

In reviewing the entire sequence of the use cycle, it is evident that transport by air in Phase III airborne operations and in administrative air deployments is within the realm of possibility and must be provided for in the design. Furthermore, movements into and out of some airports and hastily prepared air landing facilities imply trucking operations, especially where existing airports are used. Low bed trailers are one means for alleviating the restrictive dimensions in van or stake body trucks (for the latter, 7 feet is the maximum width and 22 feet is the maximum length). In the case of 12-ton semi-trailers, the allowable width that can be accommodated is increased to 8 feet, the height restriction is eliminated, and the length even without overhang can be 20 feet. The allowable weight on this trailer is 20 tons for highway operations and 12 tons under adverse conditions.

The limitations imposed by heavy air transports (C-124 and C-133) are with respect to passage through the loading doors. In the C-124, the width in the area of the loading ramp tapers down from 11 feet 4 inches at 4 feet off the ramp to a width of 8 feet 4 inches at a height of 11 feet 8 inches. The C-133 passes a cargo width of 12 feet 1 inch and a height of 6 feet 4 inches, with a nominal taper above that to a height of 12 feet. In the category of medium transports, the C-130 will pass through its rear door a rectangular unit 10 feet wide by 9 feet high. This figure appears to be a reasonable goal in gaining flexibility in air movements since it is conceivable that the latter airplane could be involved in Phase III intra-theater operations.

CHAPTER II

REVIEW OF CURRENT AND PROJECTED GEMS WITH PARTICULAR RESPECT TO THEIR STRUCTURAL DESIGN

2.1 INTRODUCTION

The vehicles reviewed and categorized in this section are drawn from those that are in operation or have been recently proposed, and are presented in order to indicate the probable characteristics and performance that may be expected from third generation vehicles. This information is considered useful as one sets out to perform a design study which includes evaluation of design criteria and approaches, and the execution of actual designs.

2.2 GENERAL DESIGN CHARACTERISTICS

2.2.1 Saunders-Roe SR N2

General Description

The SR N2 is a 25-ton to 35-ton vehicle, designed primarily to obtain operational experience, which will assist in the development of future commercial craft. The general layout is designed around a central load-carrying area to permit easy control of the C.G. regardless of the size and shape of the payload.

The buoyancy tank is incorporated so that the vehicle may float on water, and the bow shape has been determined from model tests carried out in towing tanks, to provide satisfactory hump behavior and acceptable impact forces while at sea.

Structural Aspects

Typical aircraft construction has been used in order to meet the necessary unloaded hover criteria. Materials used are high strength cladded aluminum sheets with protective coatings in keeping with normal flying boat practice.

Use of foam-filled light alloy panels in areas subjected to high noise levels is being made.

The basic structure is a composite beam formed by the boat-shaped buoyancy chamber, the deck above the air duct, and the main cargo area. Built in conformance with time-honored practice, this layout provides a rigid beam in bending and torsion.

The buoyancy chamber itself is divided into a 15 watertight bilges and also contains two 425-gallon (imperial) fuel tanks laterally located and two 100-gallon water ballast tanks for fore and aft trimming.

Cargo Compartment

A 20 foot x 16 foot cargo area has been designed to accommodate 60 seated passengers or 100 standing troops, fully equipped. Alternate loads such as two jeeps or a 3-1/2 ton truck can be accommodated. Removable side and top panels facilitate loading.

Engines and Transmission Details

Four Blackburn A129 (NIMBUS) free turbine engines, rated at 815 horsepower (maximum control) are housed in pairs in an aft engine room and drive two fans and two reversible-pitch propellers. The fans supply pressurized air for lift and the propellers supply propulsive and maneuvering thrust. Each pair of engines is coupled to one fan/propeller unit. The shafting for the forward units passes through the cargo area roof beam.

Fuel	850 Imperial gallons
Typical cruise speed	70 knots
Cruise hover height	1.0 to 1.5 feet
Maximum hover height	2-1/2 feet

Comments on Design and Operation

Although GEMs are unlikely to be commercially competitive with conventional vehicles at gross weights less than 50 tons, Saunders-Roe consider that insufficient operational experience has been accumulated to justify going from the relatively crude SR N1 to a 50-ton gross

weight machine. Consequently, the SR N2 has been designed primarily as an operational research craft for use by both military and civil operators, to obtain operational experience and to determine the limitations over water. One interesting facet of the design is the selection of fan/propeller/engine sizes based on their direct application to machines in the 50- to 100-ton gross weight size.

Experimental model tests carried out by Saunders-Roe on annular flexible trunks have now been confirmed to some degree by tests carried out on the SR N1. These trunks serve the dual purpose of reducing spray, thus increasing visibility at low speeds and increasing the maximum hover height for the same power. It can be expected that this development will be proven out on the SR N2 during 1962. When developed, Saunders-Roe see no problems to operating both ways, through breaking surf 6-8 feet high.

With the cargo floor designed to withstand loadings up to 300 pounds per square foot, a full range of military payloads can be tolerated. There are indications that the structural weights achieved on the SR N2 are close enough to provide a firm basis for projects specifically aimed at the military logistics application. However, in the LOTS carrier role with its high demand for extreme ruggedness, the SR N2 would have to be regarded as a minimal design. On this basis, one could estimate that to meet all of the design requirements for this mission, some 5 per cent to 10 per cent of the empty weight in additional "beefing up" material would be necessary.

From a transportability standpoint, the SR N2s configuration limits it to shipboard carriage, inasmuch as the structure has been designed as a homogeneous unit with no major unit breakdown.

At a selling price of some 400,000 pounds sterling ($1,120,000), a figure based on one to five quantities, the SR N2 yields a price per long ton of empty weight of $80,000, fully-equipped and ready to go.

2.2.2 VA 2 (Vickers-Armstrong - South Marston)

General Description

It is recognized that at this stage in GEM development, practical demonstration, particularly overseas, is essential with such a new type of vehicle. The difficulties attendant on transporting a large

vehicle overseas have prompted Vickers to construct a vehicle small enough to be airfreighted and sufficiently developed to prove the engineering design and the practicability of application. It is probably the only GEM at this time being developed, from the start, to be air transportable. The vehicle readily breaks down into the main body unit and six separate ducting assemblies. The two vertical stabilizers and the propulsion powerplant are also capable of rapid breakdown. This is a utility vehicle, capable of carrying four or five people, with a speed of 40 knots and an endurance of 1-1/2 hours. The vehicle has immediate application as a fast, executive transport over sheltered and inland waters and for the transport of personnel over difficult terrain where existing vehicles cannot operate.

Structure

In keeping with the Vickers aircraft background, this machine makes full use, throughout, of aircraft materials properly designed and treated against the corrosive actions of salt-water operation. The basic design of this and other Vickers' vehicles comprises a primary structure in the form of a stiff platform, taking the distributed pressure of the air cushion on the bottom surface. Fans, lift engines, and distribution ducts for the peripheral jets are mounted on this platform, with the remaining area providing cargo space or passenger cabin structure. Double curvature is reduced to the minimum and constructional details such as machined fittings are conspicuous by their absence. The primary structure is of the box beam type with bays approximately two feet apart. Transport joints are of the simple male and female spar type. Full use is made of treated, corrugated cardboard sandwiches for floorings and side panels. Simple truss mountings for the propulsion unit facilitate rapid breakdown, and the vertical stabilizers are attached with female and male union fittings of the simplest kind.

Cargo Compartment

A five-passenger seating arrangement is situated aft of the forward intake duct. Total payload is 1,000 pounds.

Engines and Transmission Details

Three Continental engines, using aviation gasoline, drive the two lift fans and separate propulsion system.

2.2.3 VA 3 (Vickers-Armstrong - South Marston)

General Description

The VA 3 or Type 3031 is the third of a series in the Vickers R&D program and probably constitutes their version of a fully-developed machine in the smaller size range. This vehicle has been designed as a true amphibian with a speed range of over 30 to 150 knots. As such, it will fit the transportation system in the role of an over-water passenger/ goods carrier operating from land bases or terminals, or for use in areas presently inaccessible to other vehicles. At a cruise height of 8 inches, a 70-knot cruise speed is anticipated.

Structure

The primary structure comprises a buoyancy tank and a ducting system which form a load-carrying platform supporting the power units and the super-structure. The materials are mainly of aluminum alloy selected from a range of aircraft type gages.

The buoyancy tank is positioned low in the structure and provides 100 per cent reserve buoyancy and stability for the vehicle when operated over water in rough weather or in the event of a breakdown.

Passenger/Freight Compartment

The passenger or freight compartment measures approximately 17-1/2 feet long by 11-1/2 feet wide and can accommodate 24 passengers or 4,200 pounds of freight.

Engine and Transmission

Four Blackburn Turbo 603 engines, using kerosene, supply lift and propulsive power. The propulsive system employs two reversible, variable pitch, four-bladed propellers. The fuel tank is a single deep tank divided to make two compartments. Unpressurized refuelling is used. Submerged pumps and cross-feeds are incorporated.

Controls

In addition to the directional control provided by the propulsive engine and the propeller system, cable-operated control surfaces on the port and starboard coamings provide effective "keel" area to prevent drift and to assist turning.

2.2.4 VA 4 123-Ton Amphibious Ferry (Type 3032)

General Description

The VA 4 is a 123-ton ground effect machine designed as a medium size passenger and car ferry, operating over economic ranges of 50 to 200 nautical miles. Used in this way, the machine can carry 200 to 400 passengers with a car deck large enough to hold 14 large cars. The operational hover height of three feet allows the machine to negotiate six-foot waves. Cruise speed is 70 knots. Buoyancy tanks are incorporated for flotation and the bow has been shaped for over-water operation.

Structure

The main structure comprises a load-bearing platform and a rigid box section enclosing the freight decks. This design enables the machine to weather rough seas and to proceed as a displacement vessel in the event of lift system component failures.

Buoyancy tanks and the peripheral ducting system are integral with the platform. The tanks situated low in the structure provide a large reserve buoyancy and stability. Strong, heavily-plated bows are fitted to withstand the impact of heavy seas and surface debris.

The superstructure is comprised of cargo holds, passenger accommodations and facilities. The upper deck, which includes the navigating cabins, is mounted over the length and breadth of the main cargo and car deck.

Fuel tanks are accommodated adjacent to the engine rooms in the buoyancy chamber.

Engines and Transmissions

Lift and propulsion engines are aircraft-type gas turbines. The lift system comprises nine 1,000 SHP DeHavilland "Gnomes"; the propulsion system, two Rolls Royce "Tynes" (TY 12) of 4,400 SHP each.

The lift engines drive centrifugal fans and the propulsion engines drive fully reversible, variable-pitch, aircraft-type propeller systems through conventional aircraft-type transmission systems.

Controls

Directional control is achieved by air rudders and propulsion system engines with 60^{o} of lateral rotation.

Roll stability is achieved by a central secondary duct running the length of the machine.

2.2.5 Design A (Detailed Project Study - now in construction)

General Description

This design is a 22-1/2-ton machine designed for research purposes; consequently, low initial cost, over-all simplicity, and long life are primary considerations. The basic configuration consists of two propulsive units in tandem, mounted on a boat-shaped hull. Two vertical fins are located at the stern. The air cushion is produced by four fans which pump air out of the bottom through a peripheral jet nozzle. Stabilizing jets divide the base in both directions.

The pilot and copilot are located forward; the flight engineer, the passengers, and the equipment are located aft. Cargo space is centrally located and measures 6-1/2 feet high by 10 feet long and 18 feet wide. The hull bottom contains 12 buoyancy chambers. Hull lines have been developed in accordance with the over-water performance in a Sea State 3 condition.

Structural Aspects

The hull consists of aluminum alloy frames, longitudinals, and stringent stiffened skirts. The structure is welded where water tightness is required and riveted in other cases. In nonstructural areas, maximum use is made of low cost materials such as waterproofed plywood and hard-molded, reinforced Fiberglas laminate. With the exception of the bow structure, which incorporates frame spaces of 15 inches, frames are located at 30 inches. Sizes vary from bow to stem as required for strength. The top of the air duct is considered as an inner deck, the top of the hull being used as the shelter deck. Compartmentation between decks is designed to provide buoyancy in the event of damage and flooding of compartments below the air-duct deck. Fuel tanks occupy the space directly under the cargo area and run to half of the width of the buoyancy compartment. A companion

way runs fore and aft on the center line of the machine. Each ducted propeller consists of an annular duct with an engine, gearbox, propeller, and cowling mounted to a center support structure.

Cargo Compartment

A 10-foot by 18-foot cargo area has been designed to accommodate 10,000 pounds of cargo at a designed overload gross weight of 27-1/2 tons. Total disposable load for both versions is 3.86 tons and 8.86 tons, respectively. A 3 foot by 2 foot hatch provides access through the upper deck.

Engines and Transmission Details

Four Solar Saturn free-shaft gas turbine engines, rated at 1,120 SHP (maximum control) are housed on top of the hull. Two units are mounted aft and drive all four fans through a mechanical transmission system. Two units are mounted in the propulsion ducts and swivel with the duct. All shafting is contained in the upper deck structure. A cross-over shaft links the two lift engines.

Controls

The vehicle is controlled as follows:

Side force is normally produced by rolling. Cushion fan speed is used to vary height. Fore and aft motion is produced by the tandem propeller pitch-power. Pitch and roll control are effected by differentially varying the fan inlet vane angles. Yaw is produced by a rudder on the rear duct. High-speed turns over water can be achieved by foils extending into the water.

Ground Handling

All maintenance, loading, and fueling may be carried out on land. Longitudinal skegs provide a level and equally-distributed loading pattern.

Principal Characteristics

Over-all length	67.5 feet
Over-all width	26.0 feet
Power plants	4 x 1,120 SHP
Fuel	1,715 U.S. gallons
Light gross weight	45,000 pounds
Heavy gross weight	55,000 pounds
Range	225 nautical miles
Typical cruise speed	70 knots
Typical cruise hover height	6 inches
Typical maximum hover height	9 inches

Comments on Design and Operation

There can be little doubt that this design is the most advanced U.S. project at the present time. As a research vehicle, it will embody the latest "state-of-the-art" know-how with contributions coming directly from research programs in the U.K. and the U.S. As a prototype, it will also be capable of growth within and without the present hull dimensions. It is anticipated that this vehicle will systematically evaluate handling qualities of GEMs, loading factors and other structural criteria, controllability, air cushion flow, duct losses, noise measurements, spray patterns, stability, and powering requirements.

It is also possible that a large number of appropriate missions suitable for the armed forces will be attempted and proven out.

While the design, particularly the design of the structure, is perhaps based on extremely conservative values, as a research machine required to operate in rough seas and other rugged environments it is

probably the most realistic GEM to be configured at the present time and is in complete agreement with the Saunders-Roe series of test vehicles.

It is anticipated the over-all cost of the prototype vehicle, which is scheduled for completion in 1963, will not exceed $2.5 million.

2.2.6 Design "B" (Detailed Project Study)

General Description

This GEM study, recently proposed, is a four-fan, annular jet machine with mixed propulsion systems. The machine has a gross weight range of 33-1/2 tons to 42-1/2 tons overloaded. It has been designed to meet rigorous military specifications as an assault and support craft.

The cargo compartment starts immediately aft of the forward fans and extends to the stern. Clearance dimensions of this compartment are 43 feet long, 20-1/2 feet wide and 10-3/4 feet high.

To facilitate over-water operation, the design is built around a twin hull concept, the hulls being designed to flying boat practice with bow shape and dead-rise. Retractable landing pads are located at the extremities of each side hull.

Welded bottom plating on each hull is incorporated to insure an excess of buoyancy.

Structural Aspects

From a detailed analysis of materials and methods of construction, the designers decided to utilize 7075-T6 as the basic structural material, and aircraft-type construction utilizing rolled hats for stiffness.

The major load-carrying hull structure is formed by the cargo deck, a forward torque box, side beams, and a twin hull bow.

External skin is stiffened by roll-formed hat sections in the longitudinal directions, on the sides, and athwartships on the center bow, where flat or single curvature panels facilitate economical fabrication on automatic riveting and spot-welding machines. This skin-stiffener

combination provides axial and shear strength for primary loading and distributes local pressure loads to frames spaced on 20-inch centers.

The cargo deck and its substructure consist of thin skins riveted to athwartship beams on 20-inch centers. Plywood panels cover the entire cargo area and are 3/8 of an inch thick. Longitudinal stiffeners on the bottom skin, which is welded, complete the structure.

The major longitudinal bending structure is made up of two side beams, the bottom skins, and stiffeners. The beams consist of the cargo compartment side walls the outer decks over the side internal air ducts, the hull sides, and the longitudinal members in the region of the side nozzles.

The twin hull bow structure is similar to flying boat design in all respects.

Cargo Compartment

A cargo compartment 43 feet by 20-1/2 feet has been designed to provide space for large, low-density items. A full width aft-loading door and ramp and full area overhead loading doors are incorporated. A 20-inch tie down grid pattern accessible through cutouts in the plywood flooring provides a flush tie down system. The deck is slightly convex to permit drainage.

Engine and Transmission Details

Two T64-GE-6 (normal rated power 2,680 ESHP) engines are mounted below the cargo floor with the engines facing aft, providing access from inside the machine for servicing, etc. In this location, one gearbox receives the power from both engines before distributing through shafting and gearboxes to the fans and propellers.

Controls

Two reversible, variable-pitch propellers are located behind the vertical stabilizers. Power is supplied as above. A twin annular jet curtain system incorporates propulsion louvers along the sides. An engine exhaust stability system divides the vehicle athwartships on the center line. This jet arrangement, together with the variable-pitch

propellers and a horizontal stabilizier, is considered adequate for all design operations. Control functions are mechanically integrated into conventional operation controls.

Comments on the Design

The most important facets of this detailed design study are:

1. Compatibility with the LOTS carrier role.

2. Component breakdown suggests a high degree of transportability.

3. Structural design is representative of an open sea operational GEM.

2.3 MATERIALS AND STRUCTURAL CONCEPTS NOW IN COMMON USAGE AND PROJECTED FOR FUTURE MACHINES

2.3.1 Basic Structural Characteristics of GEMs

The preceding review of current and projected hardware in the GEM field indicates that no hard and fast rule has yet been laid down that would characterize these vehicles in the same way that other transportation devices are recognized. Aircraft, trains, ships, and trucks all fit into well-defined patterns where no doubt exists as to what the vehicle is intended to be. Consequently, loading criteria and general structural arrangements are constant or nearly so for each type of vehicle. A case in point is the ship with its central keel, hoop frames, bulkheads, and shear decks, a design unchanged from the earliest shipping times to the present day.

The Ground Effect Machine and its structure have yet to be finally evolved, in terms of shape, size, and loading criteria, although several relatively large vehicles have been built to fit the transportation of passengers and freight over water. The widely differing philosophy varies from the view of Saunders-Roe with their boat-shaped SR N2 to the rectangular barge shape of the Vickers Type 3032. However, without exception, all first and second generation machines have utilized aircraft type, light alloy-clad sheets and box beam structures, a natural choice since their manufacturers have been largely airframe

companies. It is also apparent that future machines now in the drawing board phase will follow the same design philosophy.

At this point it is well to note why the manufacture of GEMs has and will be in the hands of aircraft manufacturers for some time to come, certainly until a breakthrough in recirculation or power plant design is achieved.

From an aerodynamic point of view the GEM is inefficient. The best L/D that can be obtained at the present time at speeds close to 80 knots is of the order of 4 to 7. Compared with the average transport aircraft L/D of 10 to 20 or the ship L/D of 100 and up, it is evident that the GEM will be extremely high-powered, with a resulting poor operational economy, unless it is possible to operate at high payload to gross weight ratios.

Consequently, the GEM's economics are more dependent upon the efficiency of the structural layout than on any other single parameter. It is for this reason that it becomes necessary to use aircraft materials and design techniques for most GEMs. Perhaps the latest developments in the field, personified by the SR N2, are some indication of where the final selection of materials and structural layouts can be found in a current vehicle. The average skin thickness in the construction of this vehicle is only .028 inch and the over-all structural weight is of the order of 17 pounds per square foot of cushion area (28% of the gross weight).

2.3.2 Strength Factors, Load Cases and Stressing Ranges of GEMs

At the present time there is considerable speculation on what loading criteria should be used in the design of GEMs. This is particularly true in the U.S., where little experience has been obtained on vehicles except as the result of extensive model test data. In the U.K., the general feeling is that loading criteria should be derived from existing aircraft practices in order to provide a firm basis for requirements which are now being considered by bodies such as the Air Registration Board and the Ministry of Aviation. A review of various companies' approaches follows.

2.3.2.1 Vickers-Armstrong (South Marston Ltd.)

General Considerations

For over-all strength assessment, the vehicle is considered in a number of representative conditions and placed in equilibrium under applied loads and reactive gravity and inertia loads. Normal, unaccelerated operating conditions provide no basis for strength, with factors of the order under consideration.

For over-water operation, it is envisaged that, in adverse operating conditions, a variety of impacts with appropriate reactive forces will occur.

In the cushion or airborne regime, this system of forces will be additive to the steady unit air and gravity loads. These combined conditions are deemed to cover emergency conditions due to engine or control failure since it is statistically improbable that adverse water conditions and engine or control failure would arise simultaneously. It is normal practice to design to lower factors in emergency conditions.

The proof and ultimate factors stated assume that all material used for stressed parts is produced to a recognized standard specification.

For wave impact cases, a trochoidal wave of height to length ratio of 1 in 20 is used. It is assumed that the GEM can operate in waves of twice the design hover height.

Crash cases considered are the craft hitting quays, jetties, or obstacles at sea at high speed, or crash-landing on the shore. In all of these instances the safety of the crew and payload were considered. The inertia forces given below are used in the design of support structures for the crew and payload and for objects in the craft that could affect passenger or crew safety if they broke loose in a crash landing.

The inertia forces in terms of ULTIMATE acceleration are:

4g down to 3g up
6g forward to 3g aft
zero to 3g sideways
MAXIMUM RESULTANT - 6g.

The local water pressures on the bow of a GEM due to wave impact depend in a critical manner on the bow angle and dead-rise angle as well as the relative speed at impact. These pressures may be readily determined when the above angles are optimized and the speed of the GEM is known.

Strength Factors

In conformity with aircraft practice, the cases for strength assessment are based, on limit conditions, that is, conditions which are considered to be of such severity that they rarely occur.

The choice of such conditions must necessarily be somewhat arbitrary at the present stage of GEM experience and could imply speed and/or water roughness limitations if the behavior of the craft shows this to be necessary.

The factors provided on "LIMIT" conditions are:

Proof factor

Ultimate factor

The ultimate factor of 1.5 is used in all cases except crash cases.

Loading Cases Considered for Current GEMs

- Wave impact
- Crash
- Beaching, jacking, and towing
- Slinging
- Mooring
- Local and general water pressures on the craft's plating
- Control system loads.

For initial stressing cases, the local water pressures on the bow are taken as 30 pounds per square inch unfactored.

All other areas of the GEM exposed to water impact are designed to withstand a water pressure of 5 pounds per square inch unfactored.

This pressure can act on sufficient area of the buoyancy platform to give the acceleration required in the wave impact cases.

From a fatigue standpoint, an operational life of 20,000 hours is used.

2.3.2.2 Saunders-Roe Ltd. (A division of Westland Aircraft Co. Ltd.)

Detailed design work on GEMs started at this company in October 1958, the result of which was the SR N1. When the design was laid down, there was very little information upon which to base the design stressing conditions. However, since the machine was primarily intended for operation over water, it was assumed that the engine failure case would provide a suitably severe design criterion. The assumptions made were that there would be a sudden engine failure when the machine was operating over waves of critical length and two feet high from trough to crest, leading to two main conditions.

The first case is the condition where the machine has been rotated by striking a wave with its stern in such a way that the maximum slamming load comes onto the bow. This leads to about an 8g acceleration on the pilot which is equivalent to a load approximately equal to 1.25 times the weight of the machine acting on the bow. Furthermore, it was assumed that this could apply to angles of yaw up to 45 degrees. This virtually stresses the attachment point of the outer rims, the engine mounting and determines such things as the fan clearance.

The second case is the condition where the machine just dives over one wave and ploughs into the next, so that a maximum longitudinal deceleration of about 1-1/2g is obtained.

From a crash standpoint, such as if the machine hits a piece of wood so that the front structure is crumpled, this gives a maximum acceleration of about 4g.

Water pressure acting on the machine is considered to be at a maximum pressure of 10 pounds per square inch on the bottom plating.

These criteria were based on models with no cushion, so that when the engine failed, the accompanying descent did not illustrate the decay effects of a cushion; consequently, the vertical rate of descent of

the inert model was much greater than would otherwise be the case.

When allowance is made for the cushion, the maximum acceleration on the bow is only of the order of 1/2g instead of 8g, indicating that the criterion is more appropriate to speeds of 90 knots than to the speeds of 30 to 40 knots consistent with the SR N1 performance characteristics. It was therefore logical that these design criteria should be used for the development of the SR N2, resulting in a primary stressing case of 12g at the bow. This acceleration is approximately equivalent to a force equal to the gross weight of the machine distributed over the bow area. As pointed out by Saunders-Roe (reference 14), this is in keeping with all forms of transports where the design stressing condition is equal to the gross weight applied at its extremities.

2.3.2.2.1 Ryan Aeronautical

A structural criterion using a minimum number of design conditions constituting reasonable maximum values for the critical loading conditions has been developed (reference 5) by this company. This criterion has been used across the board in comparing GEMs of all kinds and sizes.

During 1961 this company analyzed the machine developed by Lt. Col. J. L. Wosser and Lt. Cmdr. Van Tuyl in some detail from a structural standpoint. The specifications and performance data available on this machine were utilized.

In the development of structural criteria, Ryan assumed that the primary structure would be designed in accordance with accepted aircraft design practices.

All loads and load factors in these criteria are limit loads unless otherwise specified. An ultimate safety factor of 1.5 is applied to all limit loads.

The primary structure was designed to prevent permanent deformation at ultimate loads. The secondary structure is designed to prevent deformation at limit loads and to prevent failure at ultimate loads.

A summary of load factors used for the Ryan structural design and analysis is shown in Table 2-1.

TABLE II-1
RYAN DESIGN LOADING CONDITIONS

Component	Design Condition	Limit Load Factor n_x	n_y	n_z
Primary Structure (Basic Framework Including Bow)	Gross Weight, flight Gross Weight, landing Gross Weight, collison	+2.00 0 -3.00	± 1.00 0 ±1.33	± 1.25 + 2.00 0
Engine Pods & Supports	Maximum loads acting simultaneously	+2.00 - 3.00	 ±1.33	+ 2.00 - 1.25
Secondary Structure (Enclosures, etc.)	Maximum Loads acting separately	+ 2.00 -3.00	 ±1.33	+ 2.00 - 1.25
Crew and personnel Safety Structure	Minor crash loads acting simultaneously	-6.00	±1.00	- 2.00
Ducting, nozzles, & Air Pressure Loaded Structure	Maximum calculated Pressure	1.33 x calculated pressure		
Flotation Hull	Landing Pressure	No factor applied to calculated pressure		
Exposed Deck	Water Pressure	No factor applied to calculated pressure		

2.3.2.3 Company Design "A" (now in construction)

General Considerations

Three cushion-borne conditions have been assumed to be of major importance as far as the design of basic structure is concerned. The first of these simulates an impact with the wake of another craft at the maximum speed. The second condition corresponds to a wave impact in rough water. The third condition simulates an unsymmetrical wave impact in rough water during a turn.

The maximum bow and stern impact loadings for the second conditions were established in order to produce a load factor of approximately 8g's at the crew compartment and is in general accord with the Saunders-Roe data for the loads that can be resisted by crew members. The bow impact loads of the first condition are arbitrarily half of those for the second condition since it is intended that high-speed operation would take place only over smooth calm water with the exception of a possible impact with the wake of another craft. The bow and stern loads of condition three have again been arbitrarily established as three-quarters of those given by the second condition.

Both the second and third conditions have been fixed as general conditions; in other words, they are applicable to all speeds but the bow and stern loads are maintained constant. One specific case that the second condition will cover is a 50-knot speed in a Sea State 3. In a similar way, the third case will encompass the situation where the radius of turn is equal to 24 craft lengths at a speed of 50 knots.

Deck Loads

To Specification MIL-A-8865 (ASG). Limit floor loads for personnel floors - 300 pounds per square foot.

Propeller Support Loads

The inertia load factor requirements were established at one-half the inertia load requirements outlined under General Design Information of MIL-E-17341A. ($\pm$ 14.0 vertical, $\pm$ 11 athwartships, $\pm$ 7.5 fore and aft).

Docking Loads

A load factor of 1.0 at an impact velocity of 3 feet per second covers impacts between the craft and a dock.

Crash Loads

The ultimate crash load factors are based on MIL-A-9965 (ASG). (Longitudinal load factor shall be 20.0 and shall act anywhere within 20 degrees of the longitudinal axis. The vertical load factor shall be directed downward, normal to the longitudinal axis, and shall be equal to 10.0. The load factors shall act separately.)

Hoisting Loads

The vertical load factor of 2.0 is taken from the aircraft hoisting requirements of MIL-A-8862 (ASG).

2.3.2.4 Company Design "B" (Detailed Preliminary Design)

General Considerations

In this design, five loading conditions were considered, based on empirical considerations which conservatively cover the infinite variety of possible operating loads. They were wave impact, flotation in rough seas, ground landing, crash, and hoisting conditions.

Wave Impact

It was considered that there was sufficient evidence that a GEM would either deflect a wave crest or be deflected by momentarily increased cushion pressures as the wave was approached. Once again, the experience of Saunders-Roe was utilized and calculations were carried out to check the transient cushioning effects. To some extent the selection of wave height and GEM speeds have been governed by this work. A Sea State 3 (3 to 5 feet trough to crest), used in conjunction with MIL-A-8864 and the above work, resulted in the assumptions that the design wave impact condition would consider a 10-foot wave with a length to height ratio of 20, and that the GEM would be in a level steady flight altitude prior to contact with the water slope following the characteristic sine-shaped wave. GEM hover height at 60 knots was

established at three feet. Operations at higher speeds would be restricted to calmer water.

Rough Sea Flotation

Due to the complexity of estimating hogging and sagging loads, a simplified condition was assumed, leading to a 1.25 g load factor.

Ground Landing

Specification MIL-A-8862 specified a jacking load factor of 1.35. A lateral load factor due to small velocities during landing was used in addition to the jacking load.

Crash Loads

It was considered that the fully loaded GEM should be designed to withstand crash load factors, to prevent injury of crew or passengers. Specification MIL-A-8865 was used (8g forward, 1.5g aft, 1.5g side, 4.5g down and 2g up). These factors govern the design of the major items such as the cargo floor, engine and fuel tank structure, etc.

Hoisting

Hoist fittings and carry-through structures were designed to a limit load factor of 2g in accordance with MIL-A-8862.

2.3.3 Discussion and Summary

The Vickers, Saunders-Roe, and Ryan data on the strength requirements for GEMs are the only published statements on this subject at the present time. The first two companies are both builders of research craft that have for some time been exposed to over-water and over-land operations. In the case of Saunders-Roe, the SR N1 has shown general agreement of the estimated loads with model tests and has already accumulated several hundred hours of operation over water.

However, it would be unwise, at the present stage of development, to place too much emphasis on any one approach, regardless of the fact that operational suitability has been demonstrated to the satisfaction of the designer. With this in mind, a number of unpublished reports

were examined with respect to design philosophy, structural criteria, and general stressing aspects. It has been necessary to refer to them as Design A or B in order to protect proprietary interests. Table II-2 has been drawn to summarize the leading particulars and to provide some guidance on the transportability aspects. Modular and inflatable possibilities have been examined for each design and the likelihood of utilizing these techniques has been indicated in a qualitative fashion.

From these data there can be little doubt that two distinct philosophies are in effect at the present time, although it appears generally true that all GEMs are using modified aircraft techniques for construction and powering. The predominant philosophy appears to utilize Saunders-Roe design criteria or modifications of the main loading conditions where greater hover heights relieve the frequency of encounter of the design wave-height system. However, in some cases there is little or no provision for drop loads from maximum hover heights, a condition resulting in maximum bending of the structure. The remaining philosophy is hard to substantiate for a military vehicle, which must be rugged, dependable, and capable of long service life. In an attempt to correlate these variations, three methods of comparison have been used. The first, shown in Figure II-1, plots the gross weight of the machine against the percentage of the gross weight required for structure; structure in this case includes all primary, secondary structures, as well as the ducting where ducting constitutes an integral part of the total structure. From this figure, the differing design philosophies are evident with variations as wide as 14 per cent to 45 per cent over the size range. Over the 30- to 40-ton gross weight range, however, the range narrows from 28 per cent to 35 per cent with perhaps a reasonably good mean of 33 per cent.

In terms of the structural weight per square foot of cushion area, over the same size range the same wide variety of design philosophy is evident. Figure II-2 has been drawn for a size range in keeping with the LOTS mission. It will be seen that for five vehicles, roughly in conformance with this mission characteristic, the variation in structure weight per square foot of cushion pressure is between 5 pounds per square foot and 25 pounds per square foot.

One other attempt has been made to correlate the various designs, using the well known "Driggs" parameter for aircraft fuselages and hulls (shown in Figure II-3), where the total surface area of the enclosed structure is used to obtain preliminary weight estimates.

TABLE II-2
SIZE, PERFORMANCE AND STRUCTURAL CHARACTERISTICS
OF CURRENT AND PROJECTED GEMS

		Saunders-Roe SR N2	Vickers-Armstrong VA-2	Vickers-Armstrong VA-3
Height - feet			10-1/3	17-3/4
Width - feet		29-1/2	15	25
Length - feet		60-1/4	28-1/3	52-1/2
Speed - knots		70	40	70
Gross weight - tons		30	3-1/2	12-1/4
Range - N.M.		200	60	87
Payload - tons		10	3/4	2-1/4
Hover height - feet		1.0 to 1.5	3/4 to 1.0	3/4 to 1.0
Structure weight - %		33% approx.	45% approx.	40% approx.
Major material used		Aircraft clad Aluminum sheets	Aircraft clad Aluminum sheets	Aircraft clad Aluminum sheets
Type of structure		Flying boat hull with close pitched stringers and frames.	Widely spaced frames with box beams and trusses.	Widely spaced frames with box beams and trusses.
Design criteria		Sheltered water Operation (see text)	Sheltered water Operation and restricted overland amphibious duties.	Sheltered water Operation and restricted overland amphibious duties.
Transportability	Land	Marginal, would require special preparations.	Truck	Marginal, would require special preparations.
	Sea	Freighter deck & self ferry	Freighter	Freighter
	Air	Not possible	Transport aircraft	Not possible
Conversion capability	Modular	Propulsion Units and vertical stabilizer	This vehicle is designed on a partial module concept.	Small changes in design would provide modules.
	Inflatable	Not possible	Complete vehicle	Marginal

TABLE II-2 (continued)

		Vickers-Armstrong VA-4	Britten-Norman CC-2	William Denny D.I.
Height - feet		42	8-1/2	
Width - feet		58	17	10
Length - feet		173	27	66
Speed - knots		70	50	17
Gross weight - tons		123	2-3/4	4-1/2
Range - N.M.		250	0 - 500	
Payload - tons		35	1/2	
Hover height - feet		3.0	1.0	Water contact
Structure weight - %		43%	26% approx.	
Major material used		Aircraft clad Aluminun sheets	Aircraft clad Aluminum sheets & Plastic foam.	Sheet steel and Plywood.
Type of structure		Box beams and trusses - aircraft practice.	Lightweight aircraft construction. Large unsupported panel areas.	Boatbuilding frames, decks and bilging compartments. Side-wall keels.
Design criteria		Operation in 6 ft. seas.	Occasional water impacts in sheltered waters at 50 knots.	Inland waterway operation.
Transportability	Land	Not possible	Truck	Marginal
	Sea	Self ferry	Freighter	Freighter
	Air	Not possible	Would require breaks.	Not possible
Conversion capability	Modular	No advantage	Marginal	No advantage
	Inflatable	No advantage	Complete vehicle	No advantage

TABLE II-2 (continued)

		Design "A"	Design "B"	Ryan Aeronautical Wosser/Van Tuyl
Height - feet		24-3/4	23	19-1/2
Width - feet		22	40	40
Length - feet		67-1/2	66-1/2	68
Speed - knots		70	60 - 145	50
Gross weight - tons		22-1/2	33-1/2	31-1/4
Range - N.M.		225	200	200
Payload - tons			15 - 24	15
Hover height feet		1/2 to 3/4	3	2
Structure Weight - %		41-1/2	24%	
Major material used		6061-T6 weldable aluminum alloy in extruded sheets reinforced Fiberglas & plywood	Aircraft aluminum alloys	Aircraft aluminum alloy sheet 6061
Type of structure		Lightweight boat techniques and aircraft stressing	Aircraft construction throughout	Aircraft construction throughout
Design criteria		Rough water up to Sea State 3	Sea State 3	Sea State 3
Transportability	Land	Not possible	Requires preparation	Requires preparation
	Sea	Self ferry or tow	Self ferry	Self ferry
	Air	Not possible	Not possible	Not possible
Conversion capability	Modular	Not possible	Small design changes would be required	Small design changes would be required
	Inflatable	Not possible	Some components	Some components

TABLE II-2 (continued)

		Ford Aeronutronic LOTS Carrier	Saunders-Roe SR N3
Height - feet		17	
Width - feet		35.7	29-1/2
Length - feet		51.3	70-1/2
Speed - knots		70	80
Gross weight - tons		22	40
Range - N.M.		300	
Payload - tons		11	16-3/4
Hover height - feet		5	2
Structure weight - %		21%	33%
Major material used		Aircraft light alloys and Honeycombs	Aircraft clad Aluminum sheets
Type of structure			Flying boat hull with closely pitched stringers and frames.
Design criteria			
Transportability	Land		Marginal, would require special preparation.
	Sea		Freighter deck & self ferry.
	Air		Not possible
Conversion capability	Modular		Propulsion units and vertical stabilizer.
	Inflatable		Not possible.

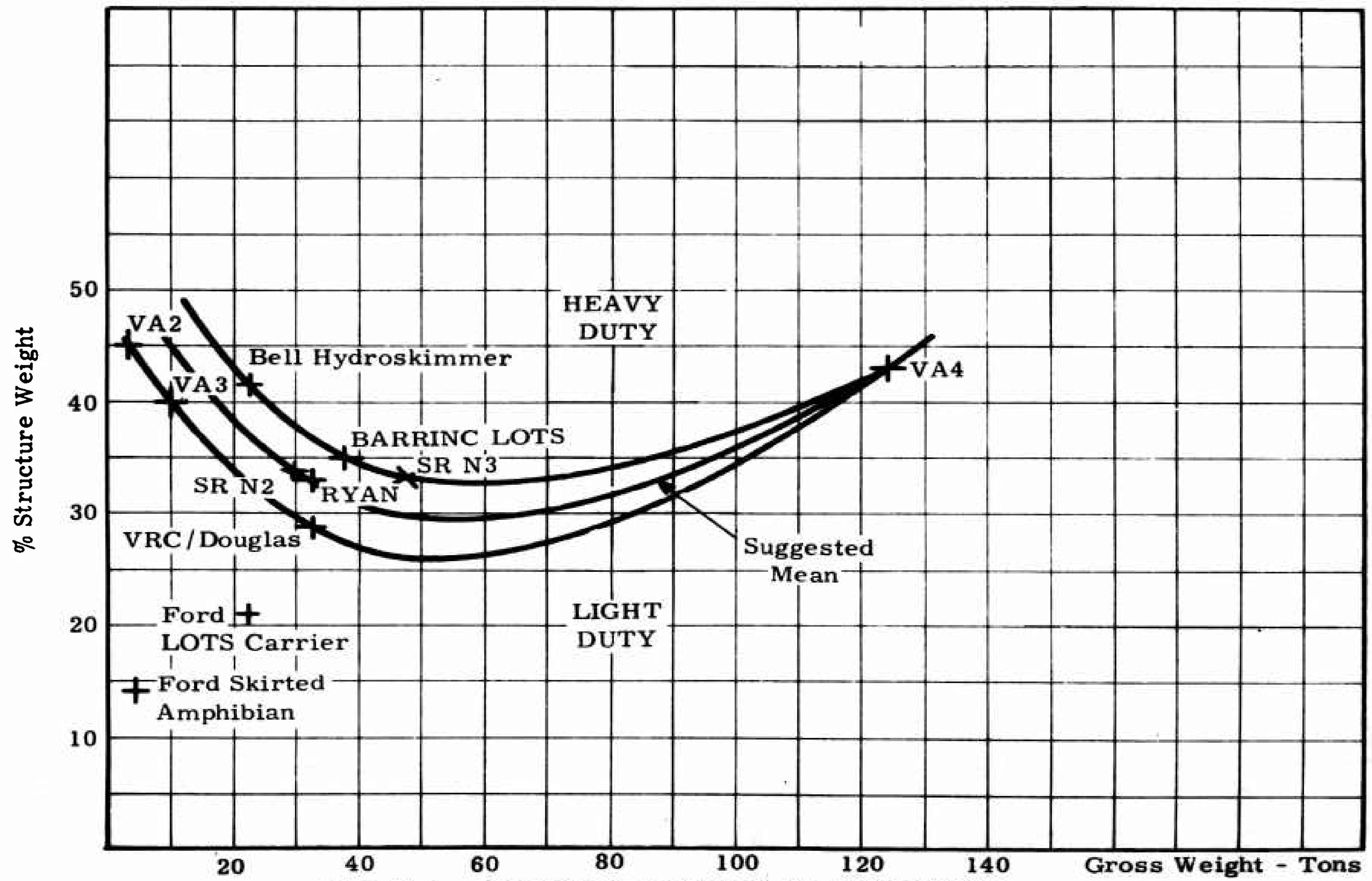


Fig. 1-1. % Of Structure Weight Vs. Total Weight

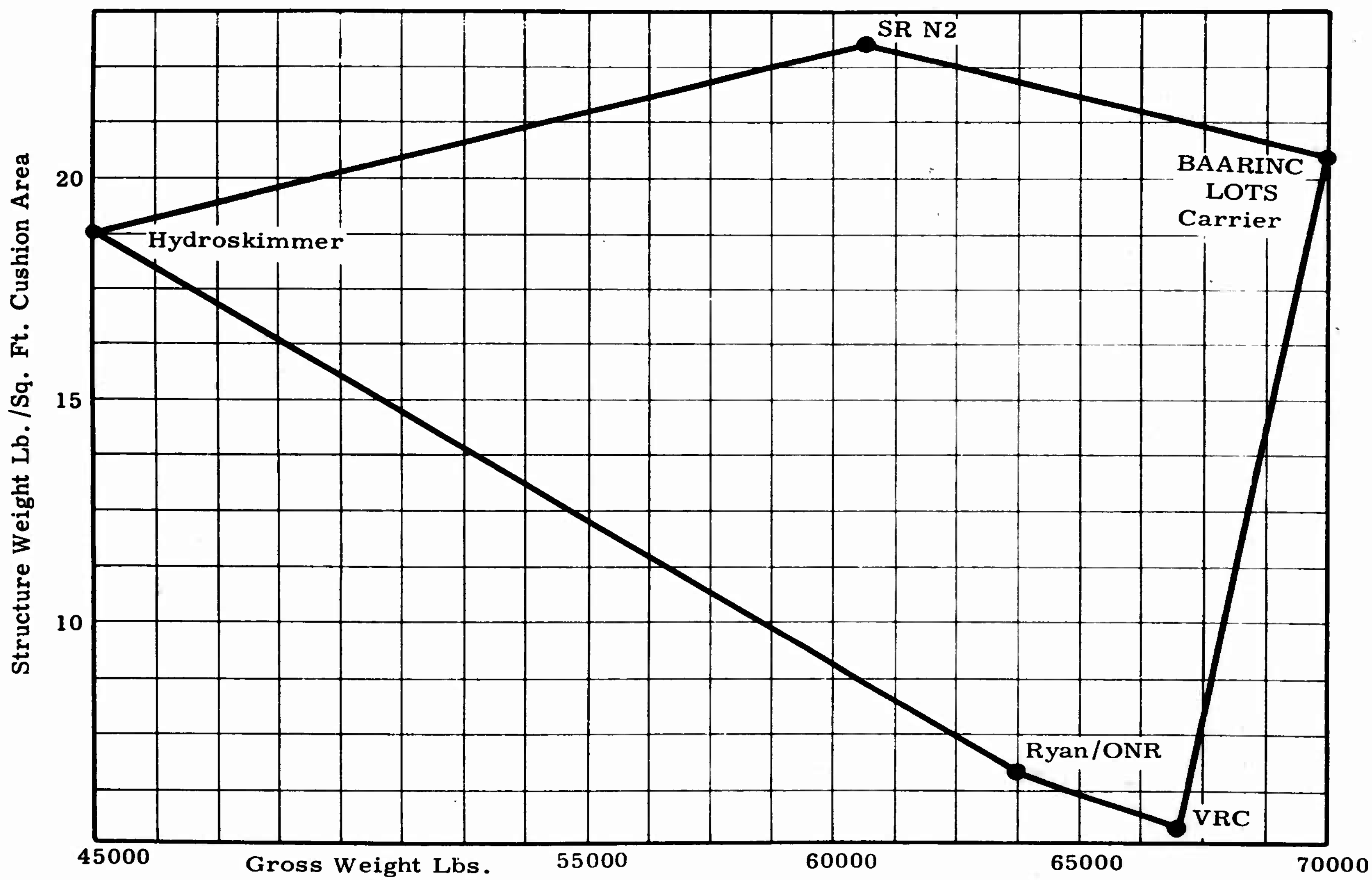


Fig. 1-2. Structure Weight Variation for Similar GEMs

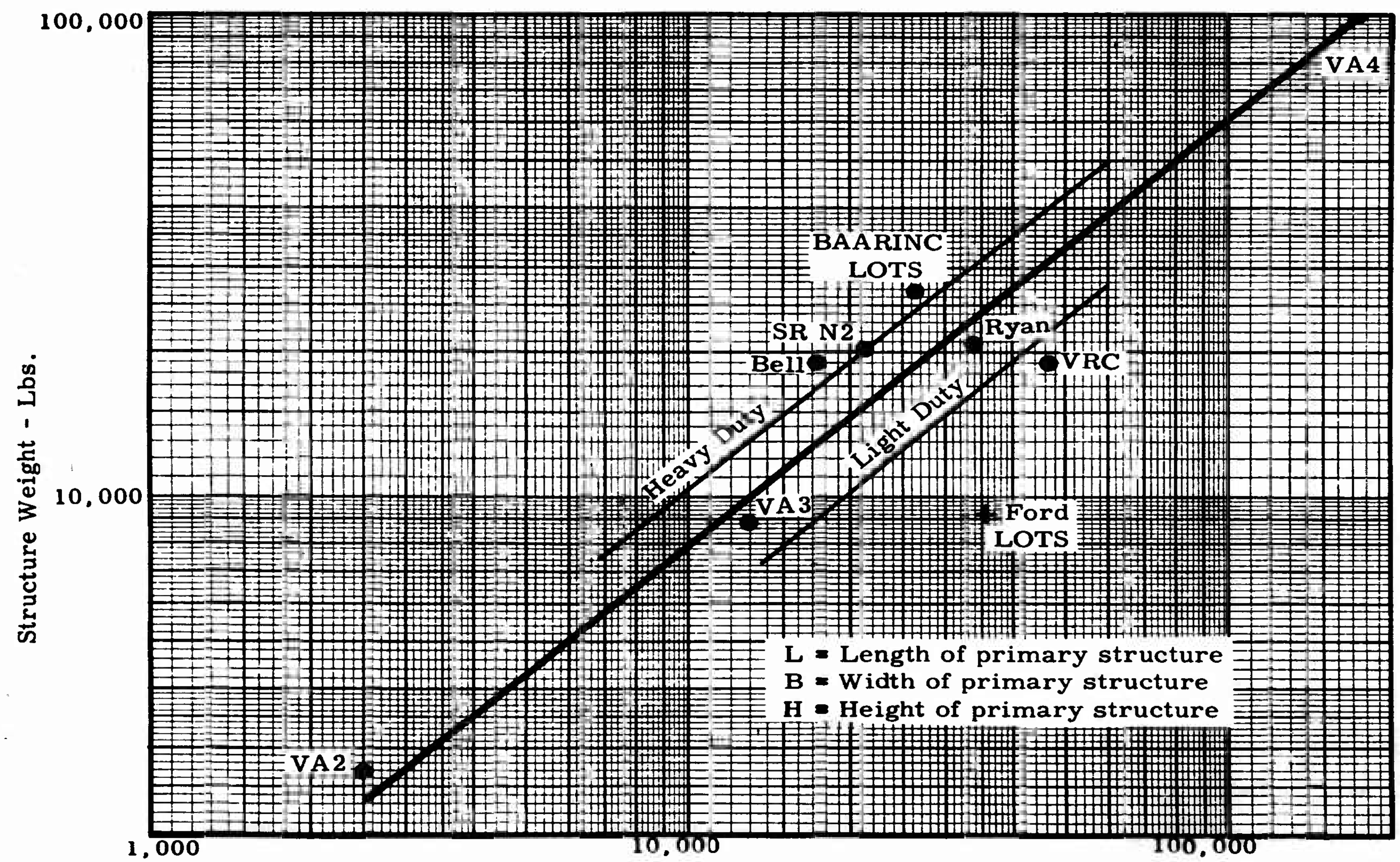


Fig. 1-3. Structural Weight Vs. Total Wetted Area Of Primary Structure

From a mean line drawn through the Vickers data, no satisfactory correlation can be expressed except to say that above the line, the vehicles may be characterized as rugged, heavy-duty type vehicles while, below the line, vehicles must be somewhat suspect, dependent upon the operational condition expressed by the designer to relieve frequency and size of load applications.

At this time, it would appear that, until service requirements are agreed to by industry, no reliable weight estimates or suitability for operations criteria can be formed.

CHAPTER III

ENVIRONMENTAL INFLUENCES

3.1 NEED TO CONSIDER ENVIRONMENTAL FACTORS

The design and operations of a military Ground Effect Machine are significantly influenced by all the elements of the total environment surrounding the operations. Military GEMs are significantly influenced by the following three types of environment:

(1) Natural environment, such as climate, terrain, vegetation, sea states, beach slopes and condition, sea approaches to coast, and natural and man-made obstacles.

(2) Induced environment, such as component-generated temperatures, noise, vibration, shock, explosive vapors, dust, spray, and exhaust gases.

(3) Combat environment, including detection, vulnerability, protection against damage, and field repairs.

A recently published BAARINC report, The Domain of the Ground Effect Machine (Reference 6), deals with the subject in a generalized manner. Material from that report and other references (7, 8) have been scrutinized, and the pertinent details which might influence vehicle design have been adapted to the purposes of this study.

The most applicable environmental regime is the amphibious operation arising directly from the role of the LOTS carrier. The requirements of marine operation are fairly well known and produce significant differences in marine equipment as compared to land-based equipment. In addition, the hazards of the military environment are known to produce marked differences in military equipment as compared to land-based equipment. In addition, the hazards of the military environment are known to produce marked differences in military equipment as compared to equipment intended for industrial and commercial use.

3.2 NATURAL ENVIRONMENTS

The natural environment affecting GEMs is nearly the same as that for all military vehicles and is primarily dependent on the geographical area of intended operations. Initial design parameters must be based on world-wide operations to the extent feasible, and the material in this report has been developed on that basis. The application of a specialized GEM, designed for a specific operational mission only, would change the relative importance of some of the extreme conditions found in world-wide operations.

The following elements of natural environment are considered to be most significant in respect to the use of GEMs:

- Temperature ranges and extremes.
- Absolute and relative humidity.
- Precipitation intensity and duration.
- Accumulated snow loads and icing rates.
- Wind speeds, including gusts.
- Atmospheric pressure.
- Solar radiation, including infrared, visible, and ultraviolet.
- Blowing snow.
- Blowing sand and dust, including particle sizes.
- Salt spray.
- Ozone concentrations.
- Fungus growth.
- Hailstorm duration and particle sizes.
- Abrasive qualities of surface materials.

- Abrasive qualities of vegetation.

- Other environmental considerations.

3.2.1 Temperature

The basic temperature range for world-wide operations is -25°F to +115°F. For operations in hot desert regions, the maximum temperature will be 125°F. For operations in arctic areas, minimum temperatures will range down to -65°F (-75°F in Greenland and the Antartic). The temperature range for shipboard operations is somewhat more limited, from -20°F to a maximum of about +100°F. The temperature range chosen should correspond to all areas of potential operation. For a vehicle such as the LOTS carrier, the range of -25°F to +115°F covers most of the areas of potential operation, inasmuch as the LOTS carrier will be able to operate only in areas open to ship navigation. Storage environments for all materials require consideration of high temperatures -- as high as 155°F -- for a period of up to four hours daily, without solar radiation. Low temperature limits for storage should be -65°F for periods up to three days. These limits apply to packaged materials and components in transportation and long-term storage.

3.2.2 Humidity

Absolute humidity ranges for world-wide operation are from .01 grains per cubic foot at 25°F to 13 grains per cubic foot at 85°F where 7,000 grains equal one pound. This corresponds to relative humidities as low as 5 per cent at 115°F and as high as 100 per cent throughout the range -25°F to +85°F. For operations in arctic areas, absolute humidity will be very low (approximately 0.1 grains per cubic foot) although relative humidities to 100 per cent are common. Humidity in the deserts may be as low as .5 grains per cubic foot at 125°F, but in some desert regions the humidity may be of the same order as world-wide conditions.

3.2.3 Precipitation

Equipment must be designed to withstand the effects of two types of precipitation -- both steady, wind-driven rain and brief torrential

downpours. Steady rain may have intensity of as much as 12 inches in 12 hours with a drop size of 2.25 millimeters predominant. These rains will be accompanied by winds up to 40 miles per hour. Torrential downpours such as are common in the tropics and in the temperate region thunderstorms range up to 7 inches in one hour with a drop size of approximately 3.2 millimeters. The peak rate is up to 2 inches in 5 minutes with a drop size of 4.0 millimeters. These downpours generally occur during the periods of calm to light winds.

3.2.4 Snow Loads and Icing Rates

Heavy loadings of snow and ice on the vehicle will not only decrease performance but will also tend to clog engine and fan intakes, ducts, and windshields. A GEM in operational use should be considered equivalent to "portable equipment" by the AR 705-15 definition. For this type of equipment the maximum snow load will be approximately 10 pounds per square foot with a density of about 6 pounds per cubic foot. GEMs which are stored in the open may receive snow loads up to 20 pounds per square foot, again at a density of six pounds per cubic foot. Inasmuch as most GEMs now being considered for vehicle application have large flat areas, the provision for snow loads may be a significant operational problem. Superstructure icing will occur in the arctic and sub-arctic (and antarctic) regions. World areas in which superstructure icing is prevalent are tabulated in Reference 6. Maximum superstructure icing rates may be as high as six inches per hour.

3.2.5 Wind

For determination of the effects of wind, consideration must be given both to the maximum anticipated steady winds and to instantaneous gusts of higher velocity. Standard wind measurements taken at 10 feet above ground level appear to be directly applicable to a vehicle such as the GEM, without modification for height effects. Inasmuch as the GEM is significantly affected by wind loads, particular attention must be given to control system requirements for handling these loads. Normal operation conditions in land areas generally include maximum winds (highest five-minute wind) of 40 miles per hour with gusts up to 60 miles per hour. In shipboard operations, wind speeds range up to 75 miles per hour with gusts of 100 miles per hour. It is presumed that during periods of such winds all operations in which GEMs could

be involved will cease; therefore, these limits would apply to vehicles which are tied down aboard ship. In storage areas and depots, maximum winds may range up to 80 miles per hour; again it is presumed that all materiel will be well secured.

3.2.6 Pressure

Atmospheric pressure ranges for GEM operations normally will be quite limited, except for air transportation. A range of 1,060 millibars to 887 millibars, corresponding to normal conditions from sea level to 3,700 feet altitude, is suitable for amphibious operations -- in fact, for 90 per cent of all overland operations in which GEMs could be used. For air transport, possible range of pressure is down to 116 millibars, corresponding to a flight altitude of 50,000 feet.

3.2.7 Solar Radiation

Maximum solar radiation corresponds to temperatures of 125°F in the desert and approximately 115°F in the moist tropics. In the desert areas, solar radiation intensity will be as high as 105 watts per square foot (360 BTUs per square foot per hour). This radiation is made up as follows: 50 per cent infrared, 44 per cent visible, and 6 per cent ultraviolet. For the moist tropics and shipboard operations, some of the radiation is absorbed by water vapor in the air so that a maximum of 90 watts per square foot may be anticipated. This includes 51 per cent infrared, 44.5 per cent visible, and 4.5 per cent ultraviolet. All materials will be subjected to long-wave radiation losses in extremely cold areas in which material surface may be cooled below the ambient air temperature. These radiation losses can be significant for long-term storage.

3.2.8 Blowing Snow

Blowing snow can clog intakes and electrical machinery as well as reduce visibility to zero-zero conditions. Even in the United States the effects of fine blowing snow have caused significant disruption in transport operations, including a recent storm in which a railroad's entire fleet of electric locomotives was demobilized. Design conditions for blowing snow should consider particles of one to three millimeters in diameter accompanied by wind speeds up to 40 miles

per hour. The velocity of the GEM over the surface may increase the relative velocity of blowing snow, sand, or dust.

3.2.9 Blowing Sand and Dust

Blowing sand in the desert regions is a significant environmental consideration because the sand can clog small openings and cause breakdown of lubricated surfaces. Blowing sand is normally limited to heights of less than five feet above the ground. Approximately one-half of the sand remains within one inch of the ground. For GEM operations the disturbance of this sand by the downwash from the jets may be an important consideration. The maximum intensity of blowing sand may range up to 10 pounds per foot cross-section. Particle sizes will range from .18 to .30 millimeters diameter, with very few particles less than .08 millimeters. Wind speeds up to 40 miles per hour at 5 feet may accompany the blowing sand. Composition of the sand is mostly quartz. Blowing dust is distributed much more evenly throughout the atmosphere. Intensity of blowing dust will range up to 6×10^{-9} grams per cubic centimeter. Particle sizes will range from 0.1 micron to 10 microns, where one micron equals 0.001 millimeter. Again, winds up to 40 miles per hour may accompany the blowing dust, although winds lower than 15 miles per hour will be most common.

3.2.10 Salt Spray

Salt spray is a significant environmental element in ocean areas and in coastal areas. Corrosion due to deposits of salt on vehicle structure and on the engine interior will greatly reduce the power plant output and structural life. Corrosion must be considered for coastal operations up to about 1,000 feet inland. In general, salinity of the oceans is approximately 3.5 per cent, that is, 35 parts dissolved materials per 1,000 parts water. Salt spray over the oceans is dependent on the wind. The intensity ranges from 4 micrograms per square meter at 10-mile per hour winds to 30 micrograms at 35-mile per hour winds, and up to 100 micrograms at 60-mile per hour winds. In surf areas, salt spray intensity may be as much as 100 times as great as over the open ocean. Salt spray in rain over water areas and near the coast may range from 2 to 20 milligrams per liter. Military experience to date has shown that the results of even the most carefully designed salt chamber tests can be misleading. The only satisfactory method of testing has been exposure of the test item to the actual environment in which use is intended.

3.2.11 Ozone

Ozone is the strong absorbent of ultraviolet radiation and causes degradation of rubber and other organic materials. Generally, ozone is formed by the photo-chemical radiation of organic pollutents. Natural concentrations of ozone near the surface range up to 3×10^{-2} parts per million, while in intense smog the concentration may exceed 5×10^{-2} parts per million. Reference 9 is the source of this data. It would appear that ozone would be too active to exist long in smog.

3.2.12 Fungus Growth

For all operations in the moist tropics and other high humidity areas the growth of fungus on nonmetallic surfaces must be considered. The standards given in military specification MIL-F-13927 (Reference 10)or in MIL-E-4970 (Reference 11) applicable to aircraft ground support equipment should be used.

3.2.13 Hail

While hail is not a common occurrence in any part of the world, the effects of a heavy hailstorm can be catastrophic for military equipment, particularly that of aircraft-type construction. Maximum concentration of hailstorms is in the mid-latitude mountains and adjacent areas, e.g., a range of 5-10 hailstorms annually in Colorado, Wyoming and Nebraska. Sizes of hailstones range from 0.5 to 0.7 inches in the United States and Western Europe, but up to 1.2 inches in India. Rare occurrences of larger hail may be encountered.

3.2.14 Abrasive Qualities of Surface Materials

Although a GEM is designed to operate without contact with the surface, occasionally during landing operations, or when encountering an obstacle, or in the event of power failure, the vehicle may scuff or slide along the surface. Abrasion of the under surface of the vehicle due to sliding on beaches, the desert dunes, and other surfaces will be dependent primarily on the hardness and sharpness of the surface materials. On a world-wide basis, beach materials are predominantly sand with rounded-to-angular particles ranging up to 2 millimeters in size. On as much as 45 per cent of coastal beaches,

larger particles such as gravel and pebbles range up to 2 inches in size (mostly rounded) and will be mixed with sand. In about only 5 per cent of the beaches will particles larger than 2 inches be encountered. For most GEM operations mud flats will be a common environmental element; however, the abrasion of vehicle under-surface in occasional contact with mud surfaces is not considered to be serious. For operation over snow, hardness of 100 to 20,000 grams per square centimeter may be encountered for particles of 1 to 8 millimeters in diameter. Hardness of wet snow is much less. The abrasive qualities of other surface materials, including coral, bare rocks and pavement materials must also be considered. It is known that coral in its growing state (below the water surface) is hard and angular, and may easily tear the bottom of a GEM which is operating as a displacement vessel.

3.2.15 Abrasive Qualities of Vegetation

Contact with coarse grasses, thorny underbrush, and rough trunks of trees and swamp vegetation may also scrape or tear the surface of the GEM. In this case damage will be due more to puncturing by sharp points than the abrasion by the vegetation material. Care must be taken in operation in mangrove swamps and other marshy areas where exposed roots, tree "breathers", and other sharp items pierce the water surface.

3.2.16 Other Environmental Considerations

For specific missions or specific geographical areas of operation, there may be other elements of natural environment which must be considered in relation to the materials used in GEMs. It is possible, for example, that in some areas insects may be a hazard, either by clogging intake screens and ducts or by chewing on materials used in construction. In most cases a specification for a particular vehicle application will include applicable environmental considerations peculiar to that use or area of operations.

3.3 THE TERRAIN PROBLEM

The simplest manner in which terrain influences can be evaluated is by relation to GEM ground clearance and other operational chacteristics.

In the present study, operating height for a LOTS carrier is determined primarily by sea conditions in which such operations are deemed feasible. Since the study includes examination of design parameters of a broader scope, it is appropriate to include the following data on the terrain-operating height function and also on terrain versus slope and speed.

3.3.1 Normal Operating Height Overland

1 foot for operations to beaches only.

2 feet for limited inland penetration.

3.3.2 Maximum Sustained Operating Height Over Water

2.5 feet for clearance of surf in 10 per cent of coast areas.

4 feet for surf clearance in 70 per cent.

6 feet for surf clearance in 95 per cent.

3.3.3 Slope Capability

10 per cent slope capability for operations over 80 per cent of beaches.

15 per cent slope capability for operations over 90 per cent of beaches, and for most near-coast terrains.

50-100 per cent slope on 10 foot banks for access from many stream valleys.

3.3.4 Operating Speeds

Limited to about 30 knots for operations in tropical coastal swamps; also limited by obstacles in the overland phase. No specific numbers can be determined without further experimental work.

3.4 INDUCED ENVIRONMENTS

The data pertaining to these environments is presented conveniently in tabular form (Table III-1). These data are self-contained and require no further comment. However, it may be observed that most of the problems associated with these environments are not of first-order in the preliminary design of vehicles and need evaluation in actual operating trials.

3.5 COMBAT ENVIRONMENTS

Tabular presentation is used again (Tables III-2 and III-3). The observations of paragraph 3.4 are applicable. In general, the construction of a GEM should produce problems similar to those experienced with army aircraft in a theatre of operations.

TABLE III-1
INDUCED ENVIRONMENTS
SUMMARY DATA

Noise

Maximum Over-all Noise Level in Decibels Relative to .0002 Millibar

Location	Acceptable (Air Force)	Experienced	Expected	Special Equipment Needed
Inside Vehicle	113	90 to 120	80 to 95	Ear protection for continuous use. Telephony for adequate operational functioning.
Close to Outside of Vehicle	113	90 to 120	90 to 120	
	See Combat Environment Summary on Noise (Table III-2)			

TABLE III-1 (continued)

Vibrations

Acceptable Limits, for crew Stations		
Frequency, (cycles/sec.)	Amplitude (in.)	Experience to date
1	$.02 < h < 2$	No indications that these values will be exceeded. No vibration problems so far.
10	$.0002 < h < .02$	
100	$.00002 < h < .0005$	

Temperatures (maximum values)

Engine	Combustion Chamber Area	Exhaust Area
Piston	240°C to 200°C	1000°C to 600°C
Jet	200°C to 250°C	800°C to 850°C

Exhaust Gases

Engine	Inert	Toxic	Water and Oxygen
Piston	77 per cent	6 to 18 per cent	17 to 5 per cent
Jet	77 per cent	1.5 to 4.6 per cent	21.5 to 18.4 per cent

TABLE III-1 (continued)

Static Electricity

Typical Accumulated Charge (volts)	Stored Energy Levels (millijoules)	Energy Level for Spark Ignition of JP-4 (millijoules)
300 to 400	2×10^{-4} to 2×10^{-2}	0.2

Effect on Personnel Bridging Gap

Initial current level -- lethal
Drops below lethal level within 2 millionths of a sec.

Solution to all Problems of Static Charge

A trailing ground contracting conductor, and good vehicle bonding throughout.

Visibility Reduction

Due to:	Comments
Configuration	No worse than a ship, better than an aircraft
Night and Bad Weather	Usual solutions applicable
Surface Disturbance	
1. Forward speed	No problem forward, upwards, or sideways. Some obstructions by surface material towards rear.
2. Hover	Severe loss of visibility. No major problem, as GEM basically stable and can move from hover location under full contact, to regain visibility.

TABLE III-1 (continued)

Ingestion

Material	Concentrations and Particle Sizes	Effect on Lift System & Propulsion	Engines
Dirt and Sand	0-80 μ 40-105 μ 105-200 μ 0.01 gms/cu.ft.	Rotor blade and intake erosion, with loss in efficiency.	Compressor blade erosion, up to 10 per cent power loss, reduction of surge margin, high turbine inlet temperatures.
Large Particles & Objects	From 200μ through 1/4 diam. bolt to 4 lb.birds	Blade and intake damage, followed by blade failure.	Nicked, dented, twisted compressor blades, damaged inlet screen facing. Stripped compressor stages, complete failure. Subsequent failure due to damage.
Ice	1/2" to 3" diam. hailstones, shaved ice	Intake damage, little or no blade damage.	Damaged intake screens. No compressor damage, occasional flame-out.
Salt Water	2 to 500 μ > 10 c.c./min./engine	Salt incrustation with loss in power.	Slight erosion, salt build-up, power loss, SFC rise.
	"Solid" or "green"	Destruction of fan blading, bending of prop blading	Flame out, inlet damage, compressor damage.

TABLE III-1 (continued)

Ingestion (continued)

Methods for Combating Effects	GEM Industry Comments
Deflectors Coatings	No indication of major problems as yet.
Debris Guards	
Debris Guards	
Deflectors, Coatings, Washdown, Inhibitors.	Indications of appreciable salt build-up in lift system intake and ducting.

Snow and Ice Accumulation

Type	Characteristic	Problem	Severity	Solution
1.	Accumulation on static vehicle in snow storm	Removal	As for existing transportation vehicles and aircraft	Standard deicing equipment, snow removal equipment.
2.	Icing at forward speeds in snow storms	Removal	As for existing transportation vehicles and aircraft	Standard deicing equipment, snow removal equipment.
3.	Operation over water in sub-freezing temperatures.	Removal	More severe than (1) or (2)	More sophisticated equipment, or new approaches.

TABLE III-2
COMBAT ENVIRONMENTS
SUMMARY DATA

Noise	
Major sound pressure levels at source.	
Engines (turbine and piston)	(12 $\log_e$ (HP) + 95 $\pm$ 10 db
Fans 4 to 6 blade, 50 to 150 HP per fan, HP speed < 1100 fps.	130 to 150 db
Over-all sound pressure levels, approximately 1 yard from complete vehicle.	
In line with intakes (propulsion, lift, etc.)	110 to 130 db
In line with exhaust	110 to 130 db
Elsewhere	90 to 110 db
Approximate spherical spreading attenuation with distance from vehicle.	
At 100 yds.	-40 db
1,000 yds.	-60 db
10,000 yds.	-80 db
Ambient noise levels	
Jungle	40 to 50 db
Coast	50 to 60 db
Quiet residential	70 to 80 db
Noisy commercial	85 to 95 db
Heavy traffic	90 to 100 db

TABLE III-2 (continued)

Noise

Approximate atmospheric attenuation with distance	Frequency (cycles per second) 10	100	1000	10,000
100 yds.	$.69 \times 10^{-3}$	3.78×10^{-3}	312.7×10^{-3}	31.2
1,000 yds.	6.9×10^{-3}	37.8×10^{-3}	3127×10^{-3}	312
10,000 yds.	69×10^{-3}	378×10^{-3}	$31,270 \times 10^{-3}$	3120

Sand and Dust

Particle Size (inches diameter)	Lightly loaded GEM 30 lbs/sq. foot		Heavily loaded GEM 80 lbs/sq. foot	
	Max. height	Max. distance	Max. height	Max. distance
.005	300h	140h	600h	310h
.01	100h	70h	300h	150h
.050	25h	20-32h	60h	42-60h
.10	15h	27-32h	35h	34-56h
.50	4h	11-22h	15h	30-60h

Note: h = vehicle operating height - feet.

TABLE III-2 (continued)

Spray and Mist

Particle Size (inches diameter)	Lightly loaded GEM 30 lbs/sq. foot		Heavily loaded GEM 80 lbs/sq. foot	
	Max. height	Max. distance	Max. height	Max. distance
.005	500h	250h	1000h	600h
.010	250h	120h	550h	270h
.050	50h	30-38h	100h	66-80h
.10	25h	21-32h	60h	44-60h

Note: h = vehicle operating height - feet.

Radar Reflectivity

Likely cross-sections for GEMs

Vehicle sizes	Radar cross-section (sq.ft.)
5 ton	5-10
100 ton	50-100

Requirements will be that vehicle radar cross-section be minimized. This is achieved by careful attention to configuration details, shaping, aspect; by shrouding propellers as much as possible; by utilizing coating media capable of high absorption at radar frequencies.

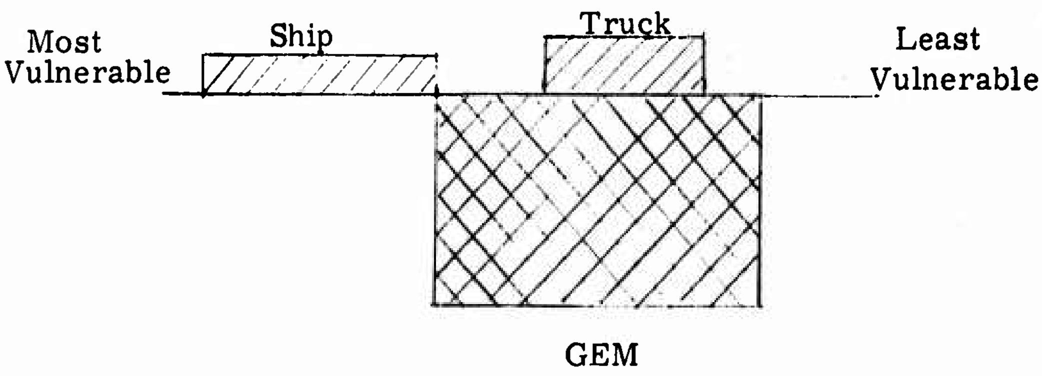
TABLE III-2 (continued)

Infrared Emanation

IR radiation - Less than 10^{-9} watts/sq. cm. at 100 yds. in wavelength range 3 to 14 microns and less than 10^{-10} watts per sq. cm. at 100 yds. in wavelength range between 1.8 and 2.7 microns.

Typical requirements are those for Light Observation Aircraft. Maximum likely radiation intensity at vehicle is 60 watts/sq. cm. at exhaust or jet pipe - rapidly reduced and dispersed by insulation, structure and atmosphere.

Vulnerability (Relative)



Damage Protection

Normal (aircraft or any vehicle) plus additional for lift system.

Field Repairs

More nearly comparable to turbine-powered aircraft than to ground vehicles. Modular construction facilitates maintenance.

TABLE III-2 (continued)

Nuclear Environment

Environment	Effect on personnel*	Vehicle*
Radiation	In hover - Intensified radioactive field -- more protection required. At forward speed - As for other vehicles of comparable speed.	In hover - As for personnel -- more careful choice of materials or more frequent inspections. At forward speed - As for personnel.
Blast pressure wave	Severe pressure fluctuations inside vehicle, unless adequate sealing and strength designed into cabin.	Uncontrollable vehicle motions if near explosion, may result in complete destruction of vehicle.
Temperature and flash	Skin burns, blindness, unless protected from high burst intensities.	Flash -- little effect. Temperatures -- engine stalling, fuel fires.

*These are predicated on vehicle being near enough to explosion for the quoted effects to occur.

Loadability at Sea

GEM limited to design operating heights of 4.5 to 6 feet; until safe unloading capabilities of cargo ships are extended to greater sea states.

<table>
<caption>TABLE III-3
SPECIAL FEATURES AND EQUIPMENT</caption>
<tr><th>Overland</th><th>Amphibious</th><th>Marine</th></tr>
<tr><td colspan="3">a. Heating and air-conditioning for crew and cargo compartments.
b. Shock and vibration isolation throughout the structure.
c. Sand, dust and spray screens or deflectors over engine and lift system intakes.
d. Engine boost capability for tropical conditions or jump capability.
e. Obstructions radar or equivalent.
f. De-icing for Arctic and Antarctic operations.</td></tr>
<tr><td>g. Anti-corrosion treatment for tropical operations.</td><td colspan="2" rowspan="2">h. Cushion edges around vehicle perimeter, for structural protection.
i. Anti-corrosion treatment, to permit sustained sea operations.
j. Flotation capability.
k. Wheeled bottom for ground handling.
l. Strengthened or protected undersurface to resist impact.
m. Hydrophobic treatment on beaching surfaces, to reduce friction and wear.
n. Clear-vision windshield.</td></tr>
<tr><td></td></tr>
<tr><td colspan="2">o. Retractable support legs for static support above rough terrain.
p. Erosion-resistant coatings in the lift and propulsion systems.</td><td></td></tr>
</table>

CHAPTER IV

LOADS AND STRUCTURAL STANDARDS

4.1 NATURE OF AVAILABLE DATA

Operational experiences for a Ground Effect Machine for the missions under consideration are not available as sources for establishing loads and structural criteria. It has therefore been necessary to combine the small number of operational data which are available, make use of experiences from related fields where they are applicable, and to fill whatever gaps remain by judicious estimates. Previous sections of this report provide the various exposures in the sequence of uses (Chapter I), current practices in the GEM field (Chapter II), and the criteria peculiar to the natural and induced environment (Chapter III). Some of the existing government requirements are applicable, both military specifications and standards (see e.g., References 12, 13). It will also be appropriate as part of this study to make some arbitrary variations in the design conditions to explore what effect will be produced on the structure and consequently its weight.

4.2 FULL-SCALE EXPERIENCES

In this category very little data has been accumulated due to the few craft operating. Systematic programs covering a variety of sea conditions, with adequate on-board instrumentation, have not been carried out. A few data appear in Reference 14, where it is reported that 4g was measured at the bow of the SR N1. There is no indication as to the composition of the motion, i.e., the heaving and pitching accelerations. Conditions under which the measurement was made are reported as 40 knots and the more usual measurement is 1.5g in 12-18 inches of chop at a clearance height (h_o) of 4 inches. The design was based on an acceleration (at the bow) of 8g multiplied by a 1.5 safety factor, or 12g condition based on ultimate strength. Peak transient water pressures between 30-40 pounds per square inch were observed. It is interesting to observe that the bottom skin of this craft, which has proved serviceable, has a thickness of .036 inches.

The resolution of a bow acceleration into pitch (α) and heave (a) components differs for all vehicles according to the mass distribution.

For the LOTS carrier a preliminary weight and balance (see Table VI-1) shows the following:

Weight (W) - 75,372 pounds

Radius of gyration (r_g) - 372 inches.

The computation for the force (F) generated by bow slamming and producing the acceleration, and the component accelerations is as follows:

$$a_{bow} = a_{c.g.} + r_1\alpha = \frac{F \cdot g}{W}\left(1 + \frac{r_1 r_2}{r_g^{\ 2}}\right)$$

where r_1 is distance c.g. to bow (use 438 inches)

r_2 is distance c.g. to force location (use 372 inches)

$$F = \left(\frac{a}{g}\right)_{bow} \frac{W}{\left(1 + \frac{r_1 r_2}{r_g^{\ 2}}\right)} .$$

The results are as follows for $\left(\frac{a}{g}\right)$ bow = 8

F = 107,000 pounds

$\frac{a}{g}$ c.g. = 1.425

$\frac{\alpha r_1}{g}$ = 6.575 (α = 5.8 rad/sec^2).

4.3 MODEL TESTS

4.3.1 Convair Tests

A few useful data points can be gleaned from Reference 15, the tests having been performed by the Convair Hydrodynamics Laboratory. The results are in non-dimensional form and for general guidance purposes they will be extended to the full scale craft dealt with in this study. Differences in the properties of the annular jets will be neglected.

Peak heave amplitude response in waves Δh/ wave height (h_w) = 1.1

associated conditions:

Frequency of encounter (f_e)/ craft natural frequency (f_n) = 0.75

Wave length (λ)/ craft length (L) = 2.00, 1.77

Wave height (h_w)/ operating height (h_o) = .98, 1.55.

Application to 15-ton amphibian

Wave characteristics, extreme - h = 5 ft., λ = 150 ft.

Speed, 40 mph = 58.7 fps; f_e = .39 cps; f_n = 0.57 cps; f_e/f_n = .69

$\lambda/L \cong 2$; $h_w/h_o \cong 2$; use $\Delta h/h_w = 1.1$ for $\Delta h \simeq 5.5$,
vertical acceleration ($\ddot{h}$) = 1.1g approximately.

Peak pitch amplitude response in waves ($\Delta\theta/\theta_w$) = 2.50

associated conditions as above;

Application to 15-ton amphibian

Speed, f_e/f_n, λ/L, h_w/h_o, Δh, all as above

Pitch acceleration (α) = 1.12 rad/sec^2

Vertical acceleration at bow due to pitching = 1.22g.

The Convair tests included several other characteristics. Models were floated in waves and the flat-bottomed craft experienced motions of the same order of magnitude as when in motion over the waves. Tests were performed to simulate power failure and accelerations on the order of 2g are reported in the fore and aft direction; no mention is made of vertical accelerations. There is no indication in any of the tests (except for the power-off condition) that the craft impacted, slammed or otherwise contacted water.

4.3.2 Saunders-Roe "Test-Computation"

The SR N1 program (Reference 14) also included some model testing. The emphasis, however, in that program was on employment of an analog computer for study of motion over waves and the main purpose in testing was determination by experiment of certain coefficients needed for analysis. On this basis the match between test results and computation are excellent over a range of speeds (variation in frequency of encounter of waves and for conditions of impact and non-impact with waves). This investigation disclosed the possibility of substantial accelerations, e.g., as follows:

	with impact	no impact
acceleration of c:g	6.0g	0.3
acceleration of bow (presumably) due to pitch and heave)	18.0g	0.4
associated conditions:		
f_e/f_n	3.5	1.5
h/h_o	1.5	0.5
λ/L	2.0	2.0

On the basis of the Saunders-Roe amplitude response results, their model is damped 7 per cent of critical in pitch and 14 per cent of critical in heave. The correlation of these computed results with the test results in the Convair tank is only fair since the lower response of the latter imply higher damping for the degrees of freedom investigated.

4.4 SEAPLANE PRACTICE

Design loading conditions employed in the seaplane art will be considered as to their applicability in GEM design. The U.'S. Navy, as part of its structural requirements (Reference 12), is concerned with water pressures acting on Vee-bottoms. For design of bottom plating and stringers and their attachment to the supporting structure, an empirical relationship between pressure and the influential factors is used:

$$p = \frac{fk\ V^2}{\tan \beta}$$

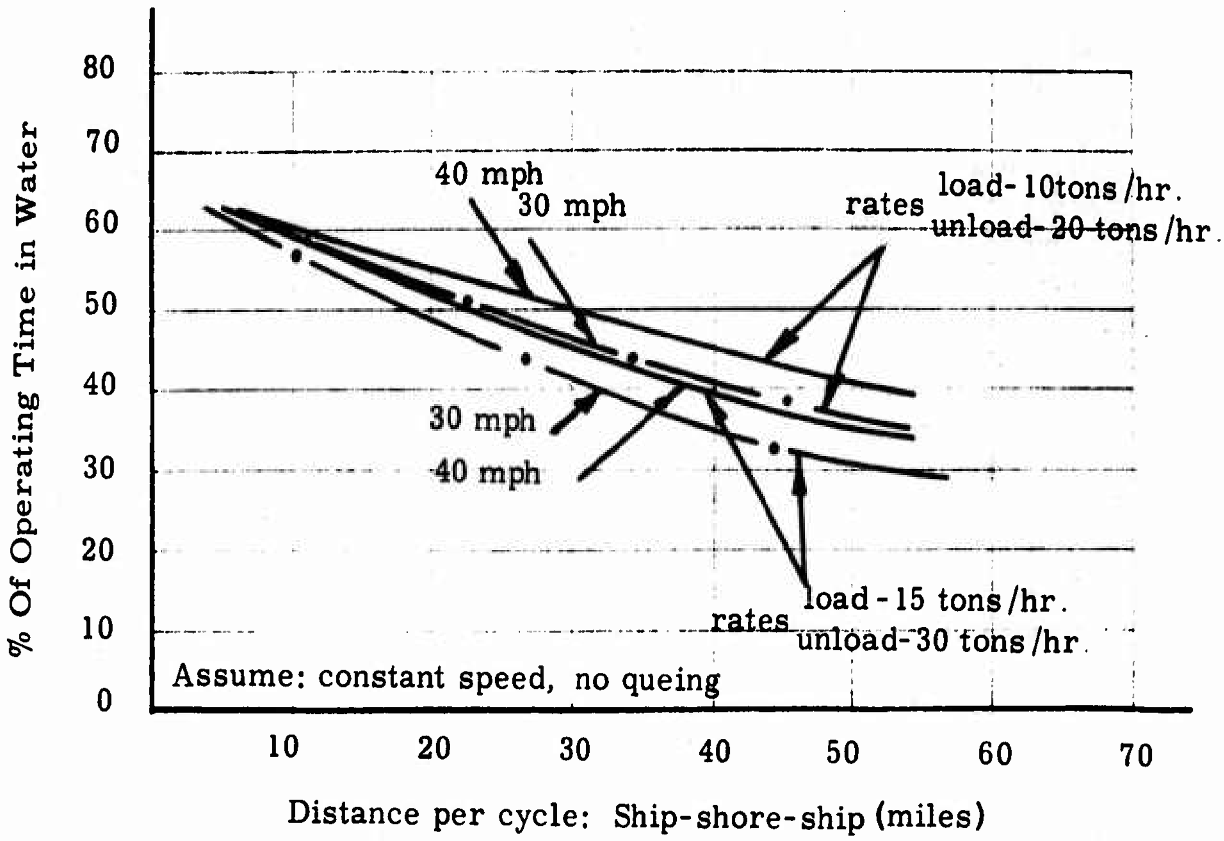
For rough water f is 0.0032, near the bow k is 2.0, and a typical dead-rise angle (β) is perhaps 15^{o}. Therefore, with V on the order of 80 knots, design pressures can be 150 pounds per square inch. The pressure diminishes in the aft direction and k is 1.0 near the step.

Another practice in this same field (Reference 16) is somewhat at variance from that required by the U. S. Navy. The reference indicates that average pressures on the order of 40-45 pounds per square inch give satisfactory designs even when peak pressures are over 100 pounds per square inch. Accelerations observed were approximately 2-1/2g on large seaplanes under correct handling, but larger values could be induced.

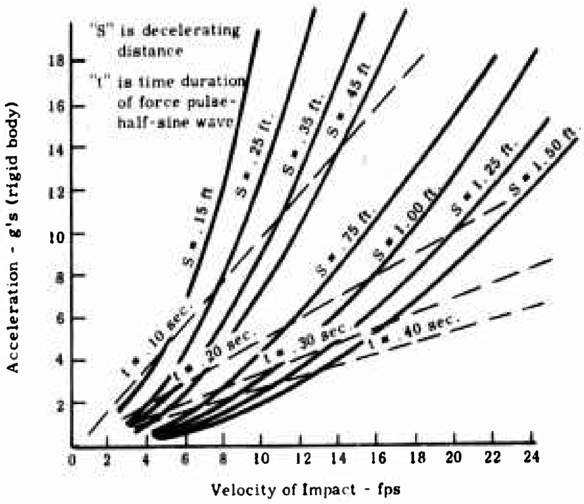
From the rudimentary comparison it would appear that seaplanes designed by U. S. Navy standards are more rugged. Operational experiences of users of both U. S. and British products tend to confirm the apparent difference. The latter are reported to be more prone to leakage and generally requiring a greater maintenance effort.

4.5 LOADING OPERATIONS AND THE ASSOCIATED CONDITIONS

LOTS operations will impose severe requirements for ruggedness on a GEM assigned that mission. The craft will be subject to the sea pounding for a proportion of its useful operating life which is inordinately high, and for practical purposes puts the vehicle in the category of marine craft. The graph below gives an indication of this. The sea action would be compounded in comparison to what is experienced by displacement craft due to the broad flat bottom dictated by ground effect machines.

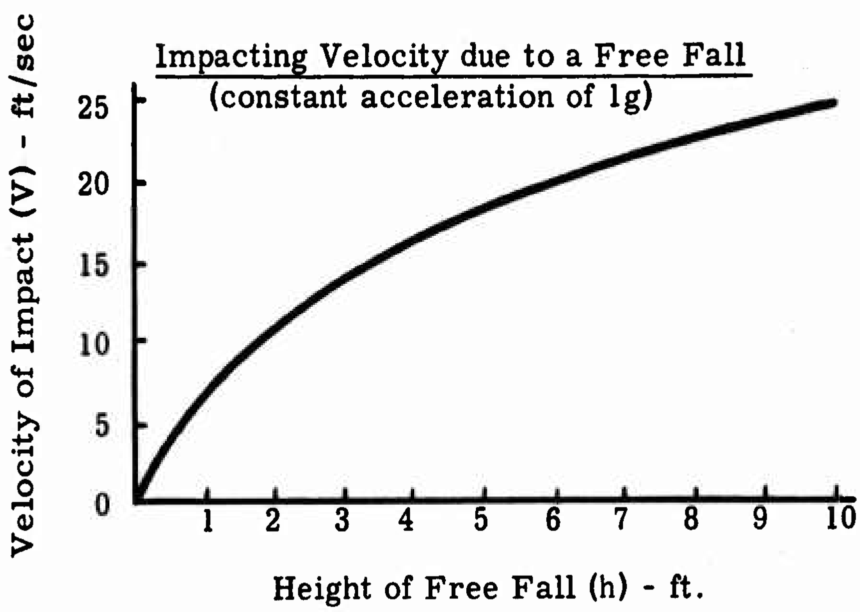


Transfer of cargo from ocean-going ships to lighters could readily cause some additional instances of high loading. Impacing of a lighter against the side of the ship will occur and must be provided for. Fenders, bumpers, and the like can be provided to cushion the impact. An example of the rigid body accelerations that might be produced can be seen by considering the following:



Impacting velocities of 10-12 feet per second are in the range of probable occurrence. Current amphibians of the Transportation Corps, which are a medium attempt at lightweight construction, also have this problem and Reference 17 indicates a need for improved fendering.

The other possibility of high impact is handling of cargo onto the cargo deck of the GEM lighter with ship's gear. The operator in lowering a load has difficulty in viewing the lighter, his signals are imprecise, and the lighter has a rapidly fluctuating component of vertical velocity. Systematic observations of the severity of this condition have not been reported, perhaps not even performed. Our estimate is that, due to adverse conditions, a load could impact the deck with a velocity equivalent to a free fall of 3-4 feet. The graph below gives the impacting velocity.



The lighter in a displacement mode or hovering, of course, provides a spring system for cushioning the impact which is softer than what could be expected in a land vehicle. If we estimate the stopping distance to be on the order of 1.0 foot, it is possible to quickly arrive at a loading condition whose approximate nature is not inconsistent with the purpose of this study. Assume a load 6 feet x 6 feet area of the base (e.g., palletized high density articles) weighing 6,000 pounds in a flat drop. From both of the preceding curves a deceleration of 5g's is determined. The corresponding deck loading is 830 pounds per square foot. The highest floor loading required in specifications and in known practice in the areonautical field is 300 pounds per square foot. Despite the significant difference, the latter figure is employed in the detailed analysis so that final design would show the influence of the few most significant parameters. Since there could be no assurance that any particular high density load could be placed on a designated part of the deck, the entire surface should be resistant to this loading.

4.6 EXCEPTIONAL HAZARDS DURING OPERATION

4.6.1 Drop Due to Inadvertent Loss of Lift

The first contingency loads the main body structure in bending. It could arise from power failure or unskilled landing on difficult terrain

and could be met by providing resistance to a drop equal to the peak operating height. Normal operating height is 2-1/2 feet, but with a reduction in cargo of 50 per cent the operating height can be boosted to 3-1/2 feet. These drops would be against a decaying cushion pressure. It is estimated that a 1-foot free drop is the equivalent of the actual drop. Estimate also that deceleration occurs through a distance of 0.33 feet; this would represent an average ground hardness. From the figures a deceleration of 6g is determined. It is assumed that landing devices are in place to transmit the loads to main structural members. In line with the approximate nature of the analysis it is assumed that a corner impact is not more severe.

A drop onto the surface of water would be a more cushioned drop so it will not produce as high a degree of loading in bending as determined above. However, surface pressures would be high and constitute a hazard. It has been shown that seaplanes are designed to resist water impacts (see paragraph 4.4) in accordance with an empirical formula where the bottom geometry is significant. The formula is applied in the analysis and produces what appear to be reasonable pressures. For example, with a dead rise angle (β) of 15°, a speed of 35 knots (40 miles per hour) and rough water operation, the pressure can be 29 pounds per square inch. To obtain pressures acting on larger surfaces (say 2-3 feet square), an average k factor is assumed at 1.2 instead of the peak of 2.0 and an additional factor of 0.6 is incorporated to account for the non-simultaneous occurrence of peak pressures. This results in 10 pounds per square inch.

4.6.2 Passage Through Surf

Passage through surf at moderate velocities can produce hydrodynamic pressures of the same order of magnitude as in the other cases of impact, but in this instance the direction is along a longitudinal axis. This presupposes operational techniques calling for a slowdown. There is no attempt at a precise determination of the motion due to cushion pressure variations over a typical surf profile but only a simplified assumption of horizontal motion impacting the craft against a stationary mass of water. If the velocity change through the surf is the order of 10 feet per second, the deceleration, referring to the rigid body accelerations during impact curves and using a decelerating distance of 1.5 feet, is 2.0g's. The pressure on a surface of 100 square feet is 11 pounds per square inch.

4.6.3 Crash Conditions

In order to improve chances of crew members and other personnel surviving a catastrophic impact (crash landing or the like), aeronautical specifications require that certain fittings and structure be designed to very high load factors. In Reference 12 the seats and their carry-through structure are identified and load factors range from 40 to 10. Higher values are specified for the horizontal (forward) direction than for the vertical (downward) and distinctions are drawn between classes of aircraft. In Reference 13 load factors for attachments and carry-through of engines, cargo, etc., are:

longitudinal	-	3.0 forward, 1.5 aft
lateral	-	1.5
vertical	-	4.5 down, 2.0 up.

4.7 SAFETY FACTORS

The widely employed factors of safety in airplane strength and rigidity determination are (Reference 12):

Yield factor of safety	-	1.15
Ultimate factor of safety	-	1.50.

While these appear very low in comparison with the safety factors in other technical areas there are several justifications. First, the loading conditions associated with well-defined operations have been intensively studied for a long period; second, the operational loads are high in comparison with the random handling and environmental loads; and, third, high quality materials and workmanship result in reduced variance in the strength properties of the vehicle.

Even though the airframe industry is the logical supplier of the lightweight structures as required for GEMs, the motivation for low factors of safety are not as strong. Consider the composition of a total factor of safety (e.g., that contained in Reference 18) in general engineering practice:

- Material factor. Strengths have variation depending on process control, inspections, behavior under differing types of loads; ratio of yield to ultimate is important for design based on yield strength.

- Load factor. Where load determination includes uncertainties a factor to compensate for this is necessary.

- Stress analysis factor. Many design problems are handled by approximate methods, some of the phenomena of elasticity and plasticity are not fully comprehended and stress results can be off in an unconservative direction.

- Fabrication factor. In the processes of construction materials can acquire changed properties and residual stresses may develop -- where these effects cannot be controlled, a factor must be included.

- Time factor. The effects to be compensated here are those that proceed with time even where the rate may vary with environment as, for example, corrosion and decay.

- Failure factor. Where the consequences of failure are dire, such as loss of life, an ample factor is introduced.

Each of the individual factors is assigned a value appropriate to the degree of uncertainty and these usually range from 1.5 to 3.0 although the failure factor in some instances can be between 5 and 10. It is thus apparent that, in general, an over-all factor of safety bears no relation to that of aeronautical practice. It is not logical in the case of the LOTS carrier design, which is more of a land-based and marine problem, to use the low value of 1.5. However, as an interim procedure -- operating data from trials being not yet accumulated -- this study will conform to the aeronautical practice.

4.8 SUGGESTED LOADINGS AND CRITERIA FOR STRUCTURAL DESIGN

The data from all the sources, and these include operations, test, and computation, indicate a few points clearly:

- Without bow slamming, the accelerations experienced by a GEM, even in rough water, will not be enough to influence structural design.

- With bow slamming, the accelerations may appear high when measured at the bow but for probable proportions the greater component of the acceleration is rotational and less severe than normal accelerations (see Figure VI-9).

- Cargo handling operations "in the stream" and exceptional hazards to the vehicle can, according to numerical engineering estimates, easily exceed the load conditions carried over directly from aeronautical practice.

To proceed with the study the following cases are selected and used for stress analysis computations:

- Longitudinal primary strength members:

  Compression and buckling - Impact of lighter against ship during cargo transfer with velocity of 10 feet per second and deceleration of 10g. Impact of vehicle with surf with velocity differential of 10 feet per second and deceleration of 2g.

  Bending - Inadvertent loss of lift and vehicle drop application of decelerating forces (226,000 pounds each) at vehicle fore and aft extremities; no safety factor in this instance. Motions in rough water induced primarily by bow slamming forces resulting in accelerations of 1.425 vertically superposed unfavorably on 5.8 rad/sec^2 pitching acceleration.

- Athwartship primary strength members:

  Compression and buckling - Impact against ship as above but use high end of range, say 12g.

  Bending - Loss of lift and hard landing reacted at positions approximately 12 feet from extremities resulting in 6g due to a 1.0 foot drop.

- Bow and bottom plating. Use 25 pounds per square inch for local, transient peak pressure on one panel generally in line with seaplane practice (at reduced speed) and also in line with actual measurements. Use 10 pounds per square inch on surfaces up to 100 square feet.

- Deck surface. Without extraordinary measures to cushion the anticipated cargo impacts, the pressure can be approximately 800 pounds per square inch. As an interim measure, use 300 pounds per square foot, an aeronautical practice carried over.

- Equipment fastening. During a catastrophic crash, the effects of which can be minimized by security of equipment tie-downs and other fastenings, accelerations of 6g are anticipated (no safety factor).

CHAPTER V

STRUCTURAL MATERIALS

5.1 COMPROMISE NATURE OF MATERIAL SELECTIONS

All of the Ground Effect Machine programs of consequence have proposed that aircraft materials be employed, namely, high strength aluminum alloys. This choice is a compromise involving many factors influencing material selection:

- strength-weight ratio, density
- ductility, malleability
- environmental resistance
- fatigue strength, internal damping
- ease of processing and fabrication, weldability
- cost, availability
- complete knowledge of properties.

Such matters as performance under extremes of high and low temperature (outside the environment to be resisted) and creep are not relevant and will not be considered.

Weight saving is of the utmost importance -- power requirements immediately attest to this. Many charts are available showing the strength-to-weight ratios and a short table (Table V-1) is included here giving the main data. There are alloy steels which are superior to aluminum even in its strongest alloys. The obvious disadvantage to high strength steels in construction is that gages would become so thin that stiffening material could quickly use up the weight saved.

There is, however, a possible weight saving if forged members, as for example in the attaching parts of the transfer joints, were to be of high strength steel. The disadvantage of dissimilar metals in the presence of the marine environment is serious and standard engineering practice would say that such usage is to be avoided. The practice of metal coating is advancing rapidly and could make possible the use of steel wherever section thickness warranted.

TABLE V-1
STRENGTH-WEIGHT DATA

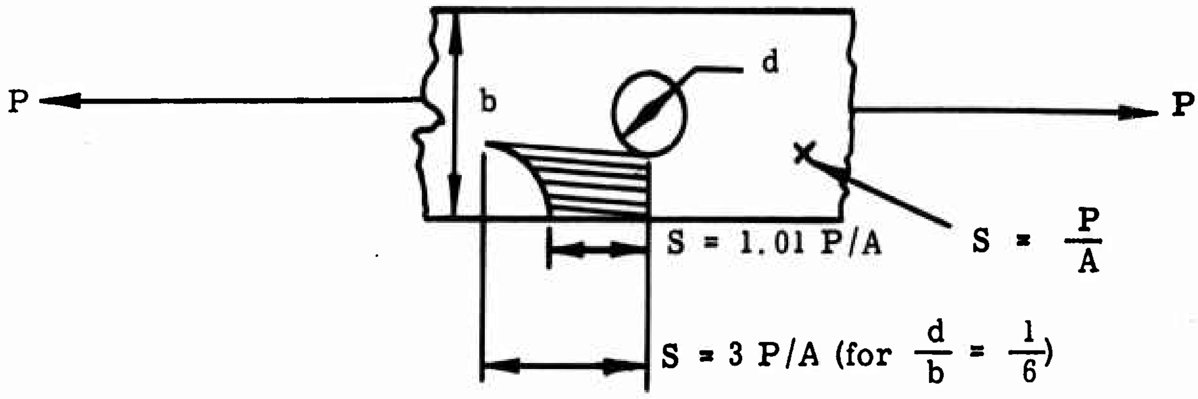
Material	(1) Yield Strength lbs/sq.in.	(2) Tensile Strength lbs/sq.in.	(3) Density lbs/cu.in.	(4) Specific Strength (2) : (3) x 10^{-6}
Aluminum Alloys				
Alclad 2014-T6	57,000	64,000	.101	0.63
2024-T4	48,000	65,000	.100	0.65
6061-T6	36,000	42,000	.098	0.43
7075-T6	67,000	77,000	.101	0.76
Steels				
.16C, .46 M sheet heat treated	44,000	61,000	.280	0.34
1025 cold finished	36,000	55,000	.284	0.19
4130, 4340 heat treated	175,000	200,000	.283	0.71
301 stainless full hard	140,000	185,000	.286	0.65
17-7 PH stainless heat treated	180,000	200,000	.276	0.72
Magnesium AZ 31B H-24	29,000	39,000	.064	0.61
Titanium 6A1, 4V annealed	120,000	130,000	.160	0.81
Beryllium 99.5%	80,000	100,000	.065	1.54
Fiberglas-Polyester		46,000	.066	0.70
Wood, spruce (2)		6,200	.015	0.41

Notes: (1) Where condition of material is not included, heat treatment to highest strength is assumed.
(2) Strength value for spruce is that used in airplane design and does not compare with values from structural design tables based on long duration peak loads; tension value of approximately 10,000 psi is not used due to great disparity between allowable bending stress and tensile strength.
(3) Data is from MIL-HDBK-5 (Ref. 19) and other sources of comparable reliability.

5.2 IMPORTANCE OF PARAMETERS OTHER THAN STRENGTH

5.2.1 Ductility

The property of ductility (the ability to yield in tension without fracture) enters into material selection and is one among several which complicate material selection. Ductility is of the utmost importance in producing safe structures, mainly by relieving stress concentrations during yielding and redistributing loads more uniformly. Consider, for example, the case of a hole in a plate or sheet as might be encountered in a riveted, pinned, or bolted joint. Standard practice would result in an estimated stress concentration factor as shown in the sketch.



As the stress passes the yield point near the edge of the hole, the performance of a material with adequate ductility is to yield without fracture and in so doing, produce a new distribution. The exact nature of the new distribution would depend on how much yielding occurred. In the case of no external load the fibers under question would actually experience a preload in compression and at some small load would be under no stress while the outer fibers were stressed positively. The above remarks are for the purpose of pointing out that ductility is a desirable property in the kind of application being considered. However, a medium degree of this quality is adequate in most of the sheet material that would be employed in the Ground Effect Machine. In many parts of the machine excessive yielding is to be avoided, e.g., the forgings of the transfer joints could be equally useless whether large plastic deformation of actual rupture

occurred. Table V-2 shows the range of values for a group of materials -- cast iron is included mainly for comparative purposes. Malleability is a similar desirable property, the difference being that it pertains to deformations in compression.

5.2.2 Corrosion Resistance

Corrosion resistance is a necessary property in a vehicle destined to operate in the amphibious military environment. Corrosion is defined as the chemical action of an environment on metals often resulting in deterioration or destruction. The essential phenomena of metallic corrosion are similar in all metals even though they occur to vastly different degrees. Side effects of corrosion are very troublesome, particularly corrosion fatigue. In this process, the endurance limit of carbon steels is drastically lowered (as much as half) from its normal value of approximately one-half the tensile strength (see Figure V-1).

Attempts to quantitatively rate the corrosion resistance of materials have not produced any useful indicators as to how any particular combination of environmental factors could be resisted. Corrosive conditions for different materials can be either an excess of oxygen (e.g., the atmosphere), water vapor, hydrogen ion and metal ion concentration or other. Results of acid immersion tests do not correlate with service (Reference 21). In Table V-3 there are comparative indications of the serviceability of a number of potentially useful materials. In general, the effect of sea water on aluminum and corrosion resisting steels is to induce local failures of the natural oxide coatings to which these materials owe much of their ability to resist corrosion. Pitting is the end result.

Prevention of corrosion is possible through precautionary measures of the following types:

- impervious coatings such as paint, vitreous enamel, resinous.

- protective metal coating which either exclude the corrosive agent or, being electronegative, corrode prior to the base metal.

TABLE V-2
DUCTILITY DATA

	% Elongation (in 2 in. length)	Remarks
Aluminum Alloys		
Alclad 2014-T6	4	Plates (this is less than sheet value)
2024-T-4	15-17	Sheets
6061-T6	10	Sheets
7075-T6	6	Thickness 0.5-1.0 in.
Steels		
1025	35-10	Annealed to hard drawn range
4130, 4340	17	Under 0.19 thickness
4130, 4340	23-10	Heat-treated, higher strengths
301 stainless	40-9	Annealed to full hard range
17-7 Ph st.	6	Sheets, heat-treated
Magnesium AZ 31B	6	Thickness Approx. .25 in.
Titanium Alloy 6A1-4V	10	Sheets, annealed
Beryllium	7-1.5	Extreme limits for alloys
Fiberglas-polyester	2-1	Varies with composition, fabric weave
Cast Iron	0.5	Approximate value (on high side)

TABLE V-3
CORROSION RESISTANCE

Material	Mfr's Rating*	Remarks-Typical Uses
Aluminum Alloys		
2014-T	C	Heavy-duty structures, aircraft trucks.
Alclad 2014-T6	A	Structures and aircraft.
6061	A	Most corrosion resistant; marine, truck and railroad applications.
7075	C	Aircraft structures.
Alclad 7075	A	Aircraft structures.
Steels		
1025 other low carbons	D	Requires protection such as coatings.
301 stainless	B	7% Ni, 17% Cr; high strength with good ductility.
316 stainless	A+	12% Ni, 17% Cr, 3% Mo; superior corrosion resistance to sea water and many chemicals.
321 stainless	A-	8% Ni, 18% Cr, 0.5% Ti, resists intergranular corrosion in welds; used for many turbine engine parts.
Magnesium	D	Alloys containing various percentages of Al, Zn, Mn.
Titanium	A	Superior resistance to sea water attack, atmospheric corrosion and sulfuric acid.
Beryllium		Resists atmospheric corrosion but only fair in presence of acids and alkalies.

*Alcoa ranking scale, arbitrary steps A-D; other ranking from International Nickel Company and Reference 20 using a scale with steps A-E.

- produce an inert coating by heat or chemical treatment of the metal surface as in anodizing aluminum.

Other precautionary measures are to select materials with regard to the particular corrosive agents present and where possible to remove the corrosive conditions. From the military preference and the actual trend of current design there is less favor to coatings, which require maintenance, and steadily increasing application of materials with inherent resistance to the environmental hazard of corrosion. Accordingly, in the final material selection, great emphasis has been placed on corrosion resistance.

There are a number of preferred design practices for the prevention of corrosion. By utmost attention to detail, the following can be accomplished:

- inhibitation of formation of electrolytic cells by avoiding dirt and moisture traps.

- crevice corrosion can be prevented by application of sealant to joints.

- attachment of dissimilar metals can be successful with gaskets and sealing precautions.

- aluminum bulkheads can take various supply lines with insulating bushings and insulating pads.

A very informative compendium of these practices along with fundamental design guidance for improved corrosion resistance may be found in Reference 22.

5.2.3 Vibration Resistance

Vibration reistance must be considered important in a highly powered vehicle. The semi-technical impression that turbine engines generate less intensive vibrations than reciprocating engines is only a subjective one due to the higher frequency spectrum of turbine vibrations. Numerous publications of environmental data illustrate this (e.g., Reference 23). Aerodynamically-generated vibrations will also propagate structural stress due to the large volume of air being pumped in the lift system. When selecting materials to be applied in a vibratory

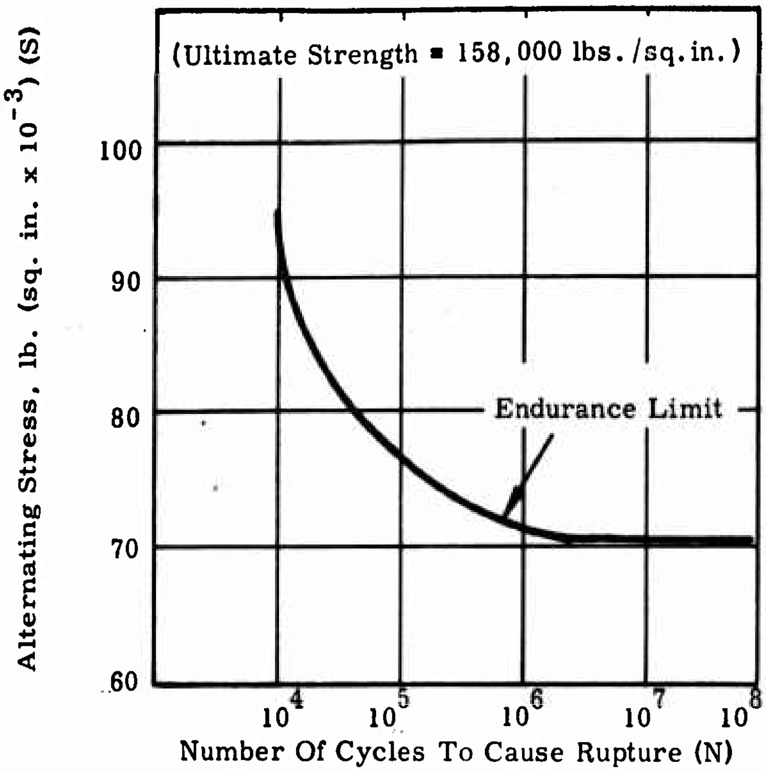
environment the main data to be considered are the S-N curves which show the decreasing stress level (S) at which failures occur for a number (N) of vibration cycles. Such curves show a definite leveling off at about 10^6 cycles for ferrous materials, as may be seen in Fig. V-1. This level is referred to as the "endurance limit" and stressing at lesser values consumes no fatigue life. Unfortunately, the endurance data for non-ferrous materials, particularly copper and aluminum, do not show as well-defined endurance limits.

S-N curves are helpful in selecting materials but they are only one of many factors to be employed in estimating fatigue life. Most of these data come from tests under laboratory conditions on reversed bending tests without any superposed steady loads, at uniform amplitude, and at low frequencies. Conditions as encountered in service can cause failure to occur at even lower stress levels (refer, for example, to remarks on corrosion fatigue).

5.3 PLASTICS

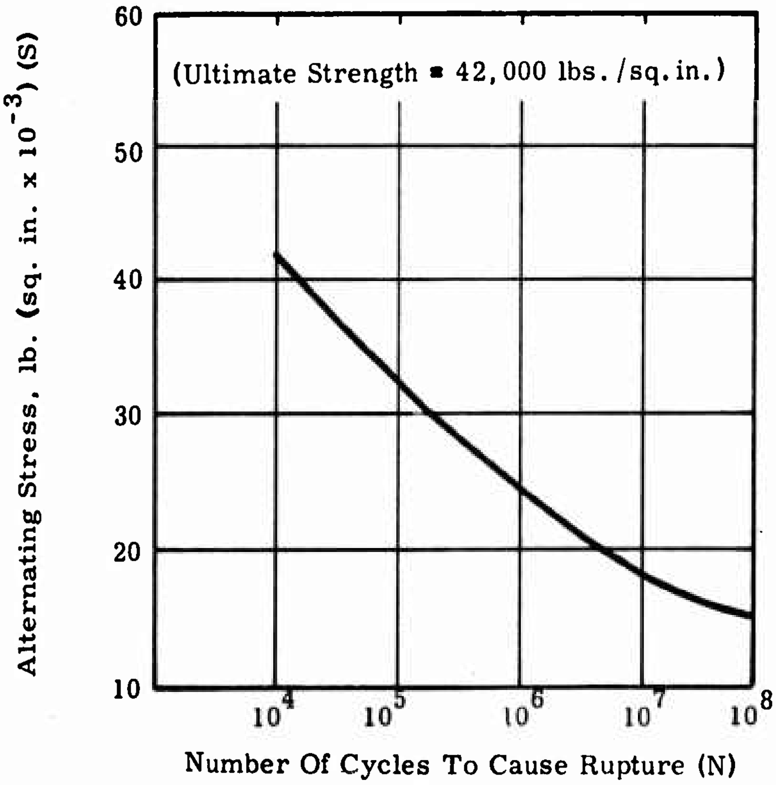
Thermosetting plastics are undoubtedly the category from which a useful structural material might come by contrast with the thermoplastic groups. The latter, including polyamides (nylon), polyethylene, and polyvinyls, among others, tend to distort under loads as light as their own weight (in structural proportions) at temperatures as low as 150° - 180°F. The thermosets, which do not soften and remelt, are in general stronger, harder, and more heat-resistant, although most are not useful above 300° -350°F. Types in this category which offer promise are phenol-formaldehyde (Bakelite, Durez, Resinox, et al), melamine-formaldehyde (Melmac), polyesters (Duraplex, Teglac). Tensile strengths are upward of 10,000 pounds per square inch, depending on the specific compositions, fillers and, for the laminates, the fabric.

Laminates of Fiberglas-polyester show excellent strength-weight properties and corrosion resistance. However, a weakness has been reported (Reference 24) in that salt water immersion causes a steady and substantial lowering of compressive strength. The effect does not show up in flexural strength and is not fully understood. This difficulty is illustrative of what can be encountered in applying new materials before properties are fully investigated. Creep, which has been considered negligible for metals, is of some importance in the case of plastics. Testing of a phenol-formaldehyde at 3,000



Fatigue Properties Of 4340 Steel (S-N Curve)
(room temperature, direct stress, unnotched specimen)

Data taken from MIL-HDBK-5



Fatigue Properties Of 6061-T6 Aluminum Alloy
(room temperature, rotating beam, unnotched, extruded specimens)

Fig. V-1. Fatigue Data For Two Materials

pounds per square inch (less than half its strength) showed for an instantaneous elastic elongation of .003 inch per inch a further elongation of .002 inch per inch after 350 hours, half of which recovered after a delay and the other half of which was permanently set. A rough comparison of creep in structural grades of plastics as against aluminum is an excess by a factor of 50. Weathering for some plastics is an undetermined property but on the whole, plastics are considered less weather resistant than corrosion resistant metals. This might appear to be a paradox since some of the same plastics are the base of protective coatings. The coatings, however, require maintenance and are not considered long-lived. Relationships between endurance limit and tensile and flexural strength are the same order of magnitude as for metals.

There is no significant advantage to be gained by employing plastic materials in the applications considered in this report. Strength-weight ratios are favorable but if the environment proves harsh to a selected plastic and its properties deteriorate, then the advantage is lost. Maintenance in service will not be minimized by selection of a plastic. Furthermore, properties of these materials have not been completely determined in the kind of service contemplated in this study. A polyester-Fiberglas combination would be the most likely candidate among plastics: fairly resistant to weathering, excellent strength, noncombustible, resistant to corrosion, moisture, fuel and oil, fungus, excellent dimensional stability. Its disadvantages are: lack of stretch (analogous to ductility), and poor abrasion resistance.

5.4 SANDWICH MATERIALS

Aluminum alloys and high strength steels are available in the form of sandwich materials. These permit the employment of very thin skin thicknesses. In some instances, the core material, which can take the form of hexagonal, square, or other cells, is a lighter material than the face; treated paper is employed in some cases. There is a need to exercise great care when employing sandwich materials, particularly that load transfer is accomplished over an adequate area. This implies extensive use of large fittings. One approach which might obviate the difficulties of joining thin skin sandwich material is the use of adhesive bonding. There is no obvious weight saving that could be accomplished by the employment of sandwich material when gages are as thick as they are required to be in most GEM designs and with the nominal proportion of structural weight devoted to stiffeners.

5.5 FABRICATION PROCESSES

Riveting is the most common joining means in fabricating aircraft-type structures. There are obvious unfavorable features to riveted designs, such as that large overlaps with a weight penalty are essential to gain space for rivets, rivet heads are dead weight in a sheared fastener, and riveted joints have stress concentrations inherently present even when all rivets fill their holes and all rivets share a total load equitably. There are two avenues for overcoming the above deficiencies -- welding and adhesive bonding. In the case of welding, there are similar difficulties for some of the stronger alloys are not readily weldable (6061 is, however, rated by Alcoa as "A" on a scale of A-D), welding of thin gages requires extraordinary skill to prevent burning and other defects, and welding of previously heat-treated material reduces the strength that is expected of the parent material although subsequent heat-treating can correct this. Adhesives are also not a straightforward solution as environmental resistance, ductility, and all relevant properties of adhesives are in some case not fully known nor fully suited to the application, and most adhesives, requiring heat and pressure during curing, are difficult to apply under usual factory conditions.

The designs evolved in this report are based on the use of riveting, which would appear to result in the most maintenance-free and reliable structure. Designs would therefore be based on a method which has been fully developed and would not anticipate any advance in the state-of-the-art. It would not, however, be a minimum weight design (see <u>Weight Saving Potential</u> in Chapter X).

5.6 MATERIAL SELECTION FOR 15-TON LOTS CARRIER GEM

To arrive at a selection of material for the main structural members, it is necessary to evaluate the conflicting factors and reach the optimum compromise. Weight-strength ratio is important in developing lightweight structures, but it is not a complete representation of the case as, for example, the use of high strength steels in applications where buckling stability results in ultrathin sheets and a maze of stiffeners. Aluminum, considering alloys and tempers with specific strengths 0.4×10^6 inches and upward, is roughly competitive with stainless steel Type 301 and similar types in the present structural application.

In the marine environment, corrosion resistance must be weighted heavily among the essential attributes of a structural material. Aluminum alloy 6061 is rated as appropriate for marine applications. It should produce the most nearly maintenance-free structure that can be achieved.

Aluminum alloy 6061 is suited to all fabrication processes that would be employed. It is weldable. Its ductility and malleability contribute to a safe structure and inhibit the development of flaws, cracks, and the like during fabrication. It is not inexpensive (at $0.54 per pound -- the 1961 quotation for flat sheet material), but it is less expensive than many other aluminum alloys and less expensive by far than some materials which appear to have superior properties almost in the "exotic" category, for example, titanium. Based on all factors aluminum alloy 6061-T6 is regarded as the optimum and is selected for the material of the main structural members.

CHAPTER VI

VEHICLE DESIGN

6.1 SYMBOL LIST

LHP	lift horsepower
PHP	propulsion horsepower
W_G	gross weight - pounds
p_c	cushion pressure - pounds per square inch
W_S	structural weight - pounds
W_{PP}	power plant weight - pounds
W_E	equipment weight - pounds
SFC	specific fuel consumption - pounds per horsepower per hour
W_F	fuel weight - pounds
$(L/D)_{eff}$	effective lift/drag ratio
NM	nautical miles
THP	total horsepower

6.2 GENERAL ARRANGEMENT

As discussed in Chapter I, the LOTS mission is essentially that of a cargo lighter; to be appropriate the vehicle needs to be strictly utilitarian -- a "work horse" type of vehicle. The utmost simplicity must pervade the design even though such a vehicle could perform other missions. Weight of all subsystems (propulsion, lifting, structural, control) must be minimized so as to permit a unit of fuel to deliver the maximum cargo. Performance capabilities beyond what

is essential -- speed, operating height above the surface, controllability, and the like -- must not add weight and subtract from the cargo which can be delivered.

The proposed vehicle layout is as follows:

- Main cargo deck, flat and uncovered.
- Lift modules flank the cargo deck.
- Annular jet and axis of lift fans are colinear where possible.
- Loading/unloading ramps, fore and aft.
- Twin bows.
- Twin propulsion units with variable pitch air propellers.

Some of the advantages in this general arrangement are obvious and others would need to be demonstrated in scale and full-size testing. The minimum ducting employed is a weight saving, while an aerodynamic penalty is questionable and would need to be a subject for testing. The independent propulsion system is simpler as compared to an integrated lift-propulsion design and hence can be expected to save weight. An advantage in transferring lift power to propulsion and vice versa is questionable. The control system is based primarily on rotation of the airscrews. A detailed investigation of these design aspects is not possible in this structurally-oriented study.

An iteration process is convenient and rapid for sizing the vehicle. Assuming initially that the highest practical cushion pressure consistent with payload density and range would lead to the most compact vehicle and least structural weight, a value of 60 pounds per square foot was selected. An initial estimate of 40 tons gross weight appeared to provide a range of 60 miles (one round-trip). This permitted size estimates and revised weight estimates until a cushion area of 1,300 square feet was selected. On this basis, a gross weight of 39 tons was forecast (see Table VI-3). (It was actually realized in the design work.) The relatively high selected cushion pressure may be considered excessive but by some authorities is considered consistent with the current state-of-the-art and highly suitable for this design. It will permit the kind of speeds required by the LOTS carrier and it will be near optimum in horsepower determination at lower speeds.

The next step was to determine that essential spaces were available within the envelope. Not only must the total surface area meet the needs but attention to proportions is necessary. Cushion efficiency is a function of length to beam ratio prohibiting this parameter from going much above 1.5, but at the same time the cargo space must be made as useful as possible. Cushion dimensions of 48 feet by 27 feet give a length-beam ratio of 1.75 and are considered a satisfactory compromise. Over-all dimensions will be greater due to the extension on each side beyond the jet center line, and due to fore and aft protrusions.

After allotting space to lifting and propulsion units, ramps, and other essentials, there remains space for a cargo deck of 46 feet by 20 feet. For the rated cargo of 15 tons, a deck loading of 33 pounds per square foot results in a very low density cargo capability as compared to the military cargo described in Chapter I. There is actually space on the cargo deck to handle three times the rated cargo at a deck loading of 100 pounds per square foot. It can thus be seen that low density cargo, which dictates the largest possible deck surface, is adequately handled by the dimensions selected.

The utility of the cargo space is influenced by its proportions. Palletized cargo and unprepared bulk articles are readily handled in the dimensions proposed. Estimating that such cargo would be 200 pounds per square foot, then 90,000 pounds (3 times the normal cargo) would occupy 450 square feet, which is roughly half of the 920-square foot cargo deck. Military vehicles which might be carried in ferry operations and lightly loaded Conex containers are the designing factors. Table VI-1 shows that the space is well proportioned for numerous combinations of vehicles. Conex containers could readily be placed two abreast in five rows for a total of ten and still leave some working space. If these gross five tons each, then the resulting cargo weight is again more than three times the rated cargo.

6.3 PERFORMANCE ANALYSIS

6.3.1 Cushion Pressure, Installed Power and Gross Weight Relationships

The following expressions are used, based on simple annular jet theory, conservative structure and power estimations, for a 2:1 planform ratio design:

TABLE VI-1
TYPICAL CARGO COMPOSITION FOR 15-TON PAYLOAD
(ARMY HANDBOOK FM101-10)

	Vehicle	Loading Condition	Number of Vehicles	Men & Matl. Add. Cargo
1	2 1/2-T LWB truck	Empty	2	2.4 Tons
2	2 1/2-T XLWB truck	Full	1	4.7
3	105 mm Howitzer	-	2	10.0
4	155 mm Howitzer	-	1	9.0
5	1/4-T Utility	Full	8	3.4
6	1 1/2-T Trailer or Water Carrier	Full	3	6.9
7	2 1/2 -T Dump truck	Empty	2	0.4
8	5-T Cargo Carrier	Full	1	0.1
9	7 1/2-T Prime mover	-	1	0.4
	Vehicle Combinations			
A	1 + 6	Full	1	3.5
B	2 + 6	Full	1	3.0
C	2 + 3	Full	1	3.7
D	1 + 4	Full	1	0.2
E	1 + 7	Empty	2	1.4
F	1 + 6 of 5	Empty	7	1.5
	Conex Containers	Full	2	4
	5-T Capacity 5.5 T Gross Wt. Full	2 @ 5-T 1 @ 3-1/2T	3	0

Total Lift HP

At 2-1/2 feet operating height, including an allowance of 15 per cent for stability:

$$LHP = 15.62\, p_c \sqrt{W_G}\ .$$

Total Propulsion HP at 40 Knots (still air)

With 45 per cent propulsion efficiency, at 2-1/2 feet operating height,

$$PHP = \left(\frac{298}{p_c} + \frac{163}{W_G^{1/2}}\right) W_G$$

where W_G is Gross Weight in tons and p_c is cushion pressure in pounds per square foot.

Structure Weight - W_S

$$W_S = [20 \text{ x cushion area (sq.ft.)}] \text{ lbs.}$$

therefore

$$\frac{W_S}{W_G} = \frac{20}{p_c}\ .$$

Power Plant Weight - W_{PP}

$$W_{PP} = (1.25 \text{ pounds per installed HP})(\text{installed HP})$$

$$\frac{W_{PP}}{W_G} = \frac{1.25 \times (LHP + PHP)}{W_G \times 2000}$$

Equipment Weight - W_E

Including crew, communications, furnishings, systems, emergency and and auxiliary equipment,

$$W_E = .05\, W_G$$

Vehicle Performance

$$\text{Range} = \frac{375}{\text{SFC}} \cdot \left(\frac{L}{D}\right)_{eff} \log_e \frac{W_G}{W_G - W_F}$$

where SFC = .8 pounds per BHP per hour.

Note: Lower figures are quoted by manufacturers, but since at this stage in the study it is not certain that the desired power will coincide with the optimum power of an available engine, a conservative SFC is chosen.

$$\left(\frac{L}{D}\right)_{eff} = \frac{2000\ W_G \times V\ \text{kts}}{\text{Total HP} \times 325} = 246\ \frac{W_G}{\text{THP}}$$

therefore

$$\text{Range NM} = 469 \times 246\ \frac{W_G}{\text{THP}} \log_e \frac{1}{1 - \frac{W_F}{W_G}}$$

Evaluation of these expressions results in the relationships that are shown in Figures VI-1, VI-2, VI-3, and VI-4.

6.3.2 Cushion Areas

With some of the vehicle dimensions determined, together with an approximate value for the vehicle gross weight, the layout can be further advanced by considerations of range and cushion pressure.

From the mission discussed at the beginning of this chapter, a range of 60 nautical miles (including an overland portion, which would be conducted at a lower altitude, particularly if prepared surfaces are available) would appear to be about adequate for the vehicle under consideration.

A range of 60 nautical miles could be cahieved, depending on cushion pressure and weight carried, by a vehicle whose gross weight lies somewhere between 35 tons and 45 tons. A preliminary estimate of the width was 27 feet, and a minimum value of the length of the cargo deck was 46 feet. From previous studies, cushion pressure of 60 pounds per square foot seems to be near optimum for this type of GEM; with this cushion pressure, the selected range is achieved with

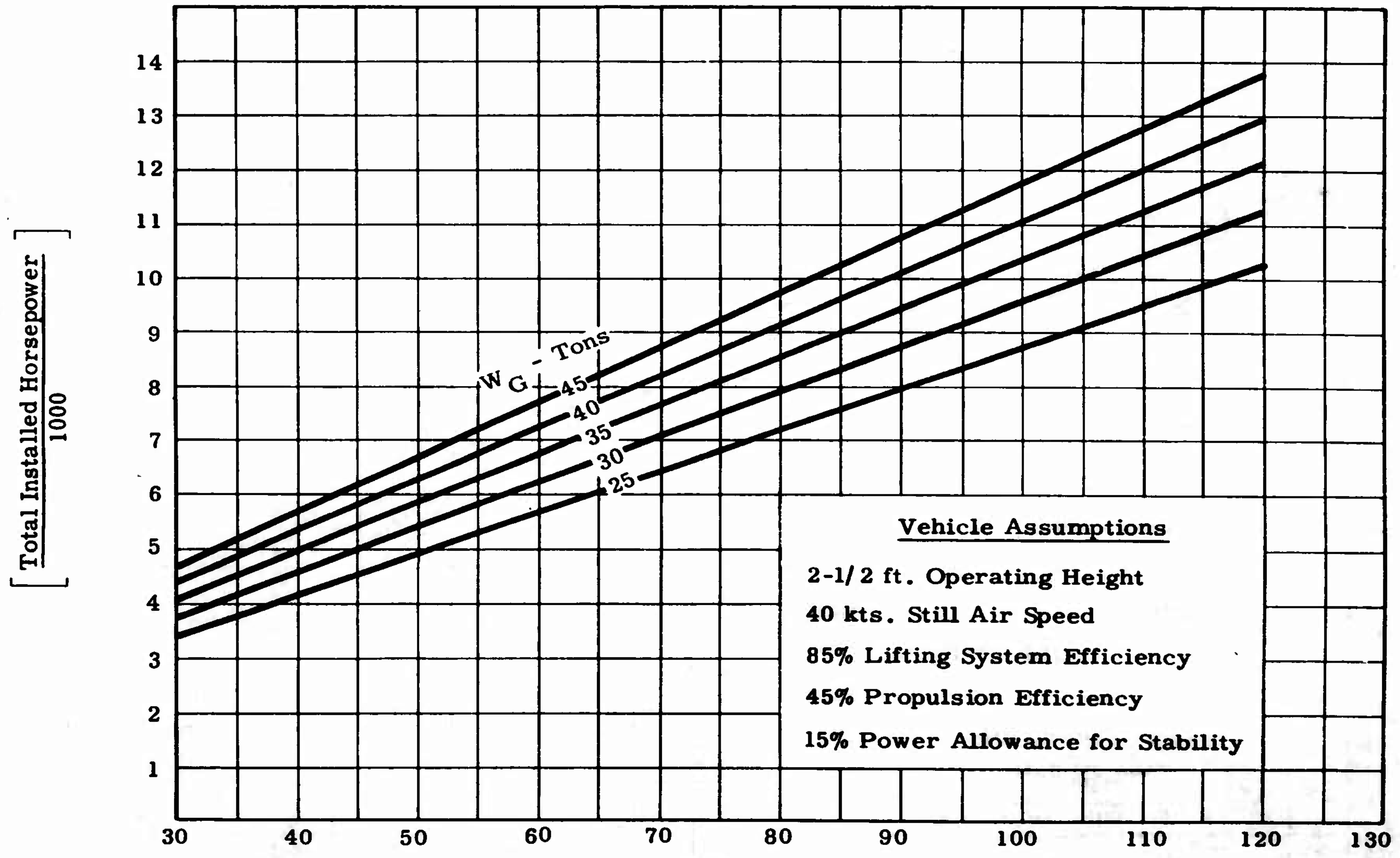


Fig. 3-2. Installed Horsepower Vs. Cushion Pressure and Gross Weight

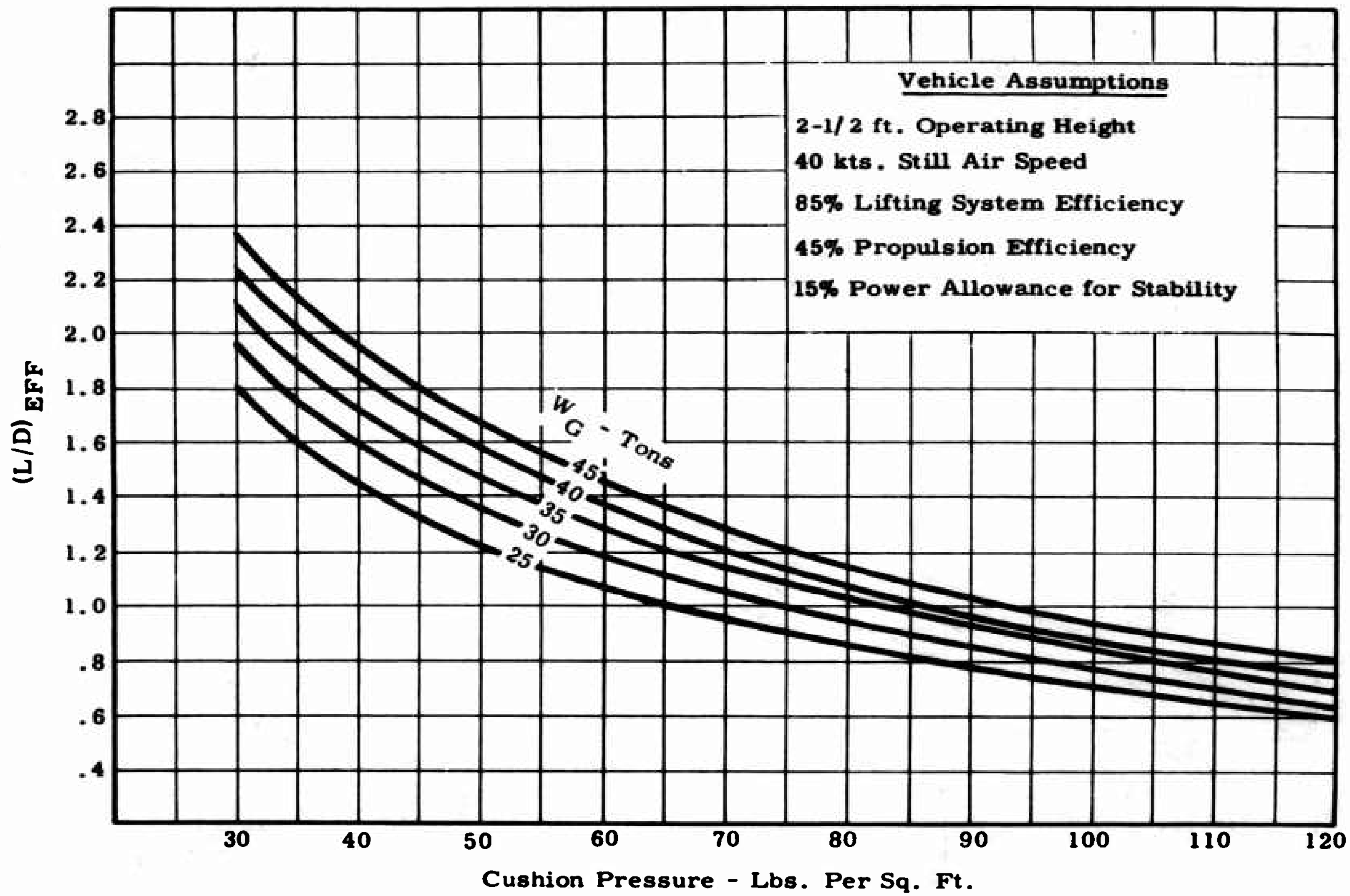


Fig. 3-3. Effective Lift/Drag Ratio Vs. Cushion Pressure And Gross Weight

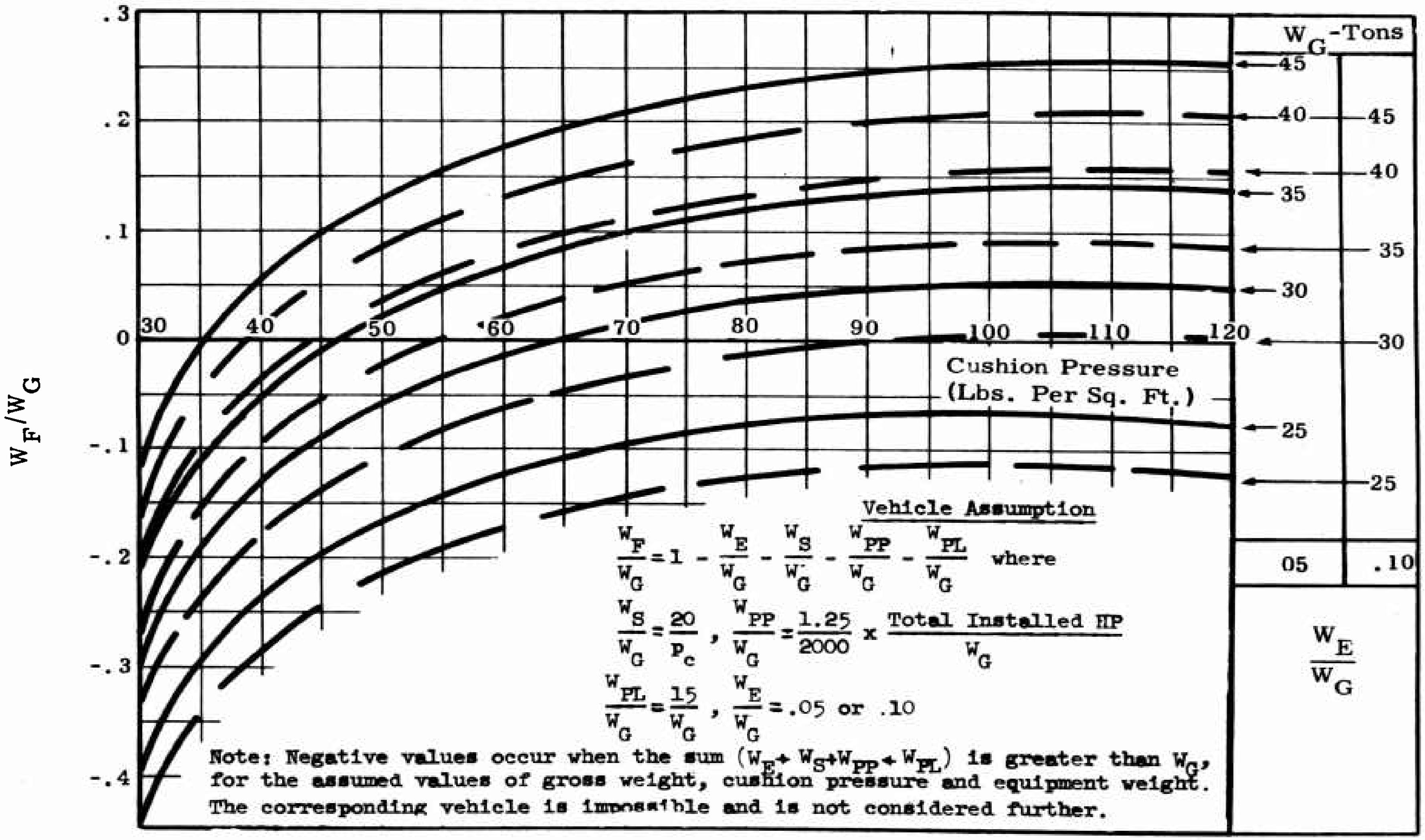


Fig. 3-4. Fuel Weight Vs. Cushion Pressure And Gross Weight

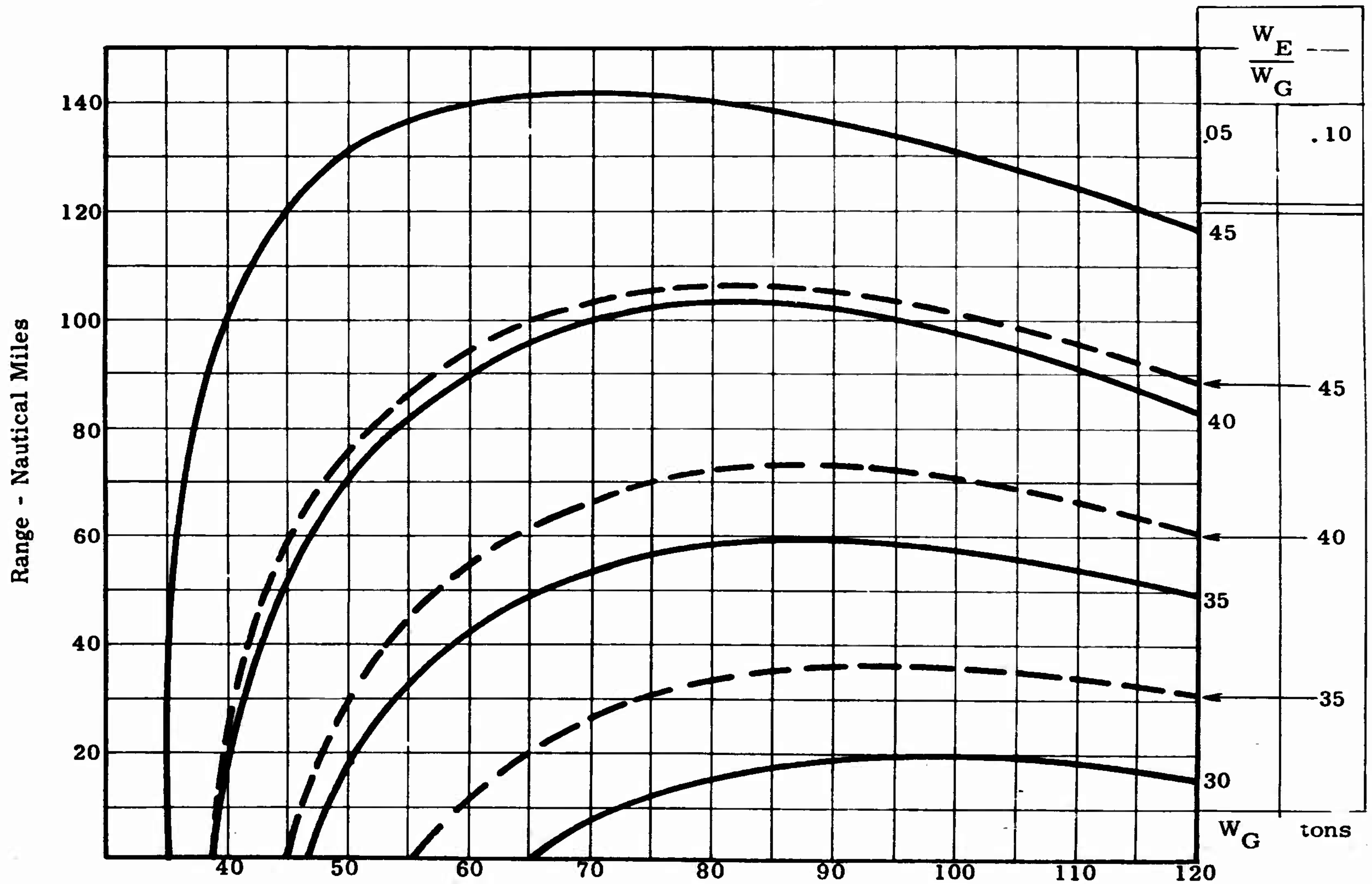


Fig. 3-5. Range Vs. Cushion Pressure And Gross Weight

a gross weight of 39 tons and a cushion area of 1,300 square feet. This results in a cushion perimeter 27 feet by 48 feet (150 feet in length), which is close to that required by preliminary analysis.

6.3.3 Power Plants

From Figures VI-1, VI-2, VI-3, and VI-4, using the cushion pressure and gross weight determined, 5,860 horsepower for lift and 1,220 horsepower for propulsion will be required. It is considered that a satisfactory layout will utilize six lift engines and two propulsion engines.

In the overload condition, when the airjets and fans are operating "off design", the gross weight will be 78 tons, at a cushion pressure of 120 pounds per square foot. Under these conditions, for the same power supplied to the fans the operating height will be approximately six inches, assuming off-design fan operation and increased duct losses.

6.3.4 Fan Size

Fan size is determined by utilizing the optimization methods of Reference 25; effects of ram recovery will be ignored. Using the previously determined values of

cushion area	$S = 1{,}300$
cushion perimeter	$C = 2(48 + 27) = 150$
operating height	$h = 2.5$

$$\frac{Ch}{4S} = \frac{150 \times 2.5}{4 \times 1300} = .0722$$

from Reference 25, jet thickness = .25 h; then

$$\frac{C\, t_e}{4S} = .01805.$$

For this value of $\frac{C\,t_e}{4S}$, and $p_c = 60$, and augmentation efficiency $n_a = .8$ and $S = 1{,}300$, the volume flow Q is obtained from

$$\frac{(n_a)^{1/2}\,(Q/S)}{p_c^{1/2}} = 2.13,$$

i.e.,

$$Q = 24{,}600 \text{ cubic feet per second.}$$

Similarly the total head at the jet exit, H_e, is obtained from:

$$\frac{\eta_a \left(H_e - p_o\right)}{p_c} = 1.51,$$

i.e.,

$$H_e - p_o = 113 \text{ pounds per square foot.}$$

The ideal horsepower, HPi, is obtained from:

$$\left(\eta_a\right)^{3/2} \left(\frac{HPi}{W}\right) \frac{103}{(p_c)^{1/2}} = 5.9,$$

i.e.,

$$HPi = 5{,}060.$$

The fan characteristic is given by

$$\frac{\sigma}{n} = 2.105 \left(\frac{Q}{N}\right)^{1/2} \left(\frac{H_e - p_o}{\rho}\right)^{-3/4}$$

therefore

$$\frac{\sigma}{n} \quad .04460.$$

For a range of ω (rotational speed in revolutions per second), the following total lift HP's are obtained:

ω	10	30	50	70
HP	5,650	5,940	6,150	6,320

Of the engines examined (see Chapter VII) the "Gazelle Junior" comes closest to meeting this requirement, since it has an output speed of 50 revolutions per second and a continuous HP rating of 800. Hence, six "Gazelle Junior" engines are considered in this design. Although at their current state of development they would result in a slightly underpowered vehicle, it is considered that future development would made up this deficiency.

Having selected the revolutions per second at which the fan is to operate, the optimum fan diameter is selected from

$$\frac{D_{opt}}{\left\{1 - \left(\frac{D_H}{D_{opt}}\right)^2\right\}^{3/7}} \cdot \left(\frac{30}{H_e - p_o}\right)^{1/2} = 2.7$$

where D_H = hub diameter $\doteq$ 1/3 D_{opt}.

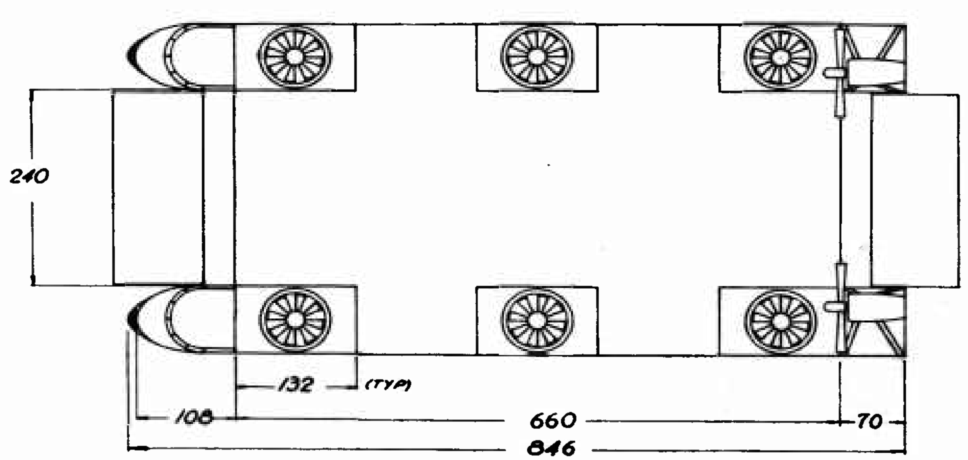
This results in D_{opt} = 5.5 feet for this vehicle. As the details of the fan design are not important to this study, they are not developed here.

6.3.5 Over-All Dimensions

Having decided on the cargo area dimensions, the cushion dimensions, the jet thickness and angle, the power plant requirements, and the fan size, the over-all vehicle layout is determinable. A preliminary sketch of the vehicle plan is shown in Figure VI-5.

CHARACTERISTICS

LENGTH (RAMP RETRACTED)	70.5 FT.
WIDTH	34.0 FT
HEIGHT	21.0 FT
WEIGHT EMPTY	40464 LB
A.U. WEIGHT	78044 LB
CUSHION AREA	1296 FT^2
CUSHION PRESSURE	60 LB/FT^2
SPEED (MAX)	40 M.P.H.
RANGE	60 M.
OPERATING HEIGHT	2.5 FT
POWER (LIFT)	5400 SHP
POWER (PROPULSION)	1750 SHP
CARGO (NORMAL)	15 TON
CARGO (OVERLOAD)	45 TON
CARGO AREA	1100 FT^2



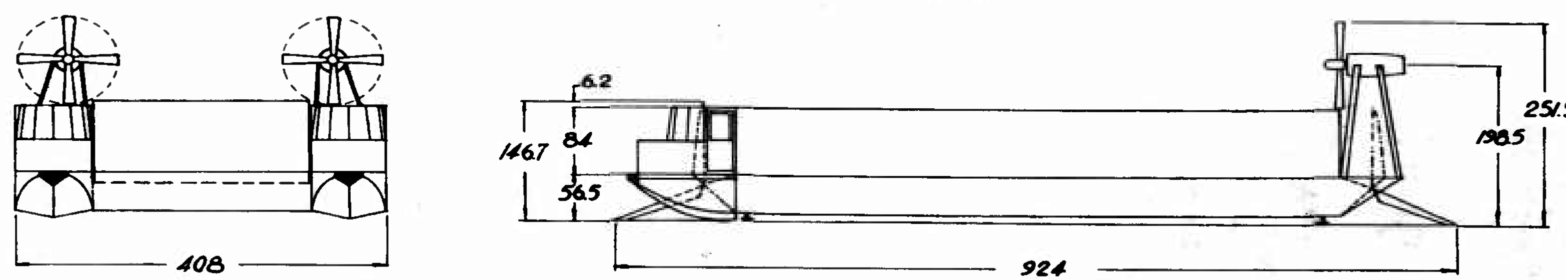


FIG V-5 GENERAL ARRANGEMENT OF VEHICLE

6.3.6 Maneuvering Capability

6.3.6.1 Side Forces

All available side thrust on one side will provide the maximum maneuver g's in the lateral direction, neglecting aerodynamic side forces when the vehicle is sideslipping.

On the assumption that all modules will produce the same thrust, the front and rear modules may have different louver geometry when compared to the center modules.

$$\text{Total maneuver g's} = \frac{\text{total side thrust}}{W_G}$$

$$= \frac{3 \times T_L}{W_G}$$

Louver thrust,

$$T_L = \eta_L \rho Q_L V_L = \eta_L \rho Q_L \sqrt{\frac{H_L - p_o}{1/2 \rho}}$$

$$= \eta_L \sqrt{2\rho (H_L - p_o)}$$

Louver area

$$A_L = Q_L \left[\frac{1/2 \rho}{H_L - p_o} \right],$$

where

Q_L = volume flow through louver

V_L = velocity of flow through louver

H_L = total head of flow through louver

η_L = louver efficiency factor.

$$T_L = \eta_L \rho \frac{(24,000 - Q_E)}{6} \sqrt{\frac{H_E - p_o \quad 1/2 \Delta H_D}{1/2 \rho}}$$

where Q_E and H_E are the values required to maintain the desired operating height, ΔH_D is the loss in total head in duct between the fan and the annular jet.

Therefore,

$$T_L = \eta_L \sqrt{\frac{2\rho}{6}} \left(24,000 - \frac{[V] \times 1300 \times (60)^{1/2}}{(.8)^{1/2}} \right) \sqrt{[H] \times \frac{60}{.8} + 1/2 \Delta H_{LDL}}$$

where [V], [H] are the volume flow and total head parameters from Reference 25.

Therefore,

$$T_L = \eta_L (2430 - 1140 [V]) \sqrt{[H]}$$

and

$$A_L = \frac{15.68 - 7.34 [V]}{\sqrt{[H]}} .$$

The results of this analysis are developed in Table VI-2, and show that this version will have poor lateral maneuverability by virtue of its low installed power.

6.3.6.2 Braking and Accelerating Forces

Maximum accelerating force will be at zero forward speed; with the contemplated propeller design this will be approximately .065g's.

Maximum braking force will be at maximum forward speed of 40 knots and will be approximately .095g's.

In actual practice it may be necessary, to supplement both of these capabilities by deflecting the thrust from the side louvers in a fore and aft direction. Average values over the speed range are:

TABLE VI-2
MANEUVERABILITY ANALYSIS

$\frac{t_e}{h}$	.25	.35	.45	.55	∞
h	2.5	1.788	1.39	1.138	0
H	1.51	1.29	1.17	1.10	.96
[V]	2.13	1.75	1.55	1.30	0
2430-1140 [V]	0	430	660	950	2430
T_L/η_L	0	489	714	996	2380
15.68-7.34 [V]	0	2.83	4.28	6.13	15.68
A_L sq.ft.	0	2.49	3.96	5.84	16
Total Sideforce lbs. (η_L = .8)	0	1170	1710	2390	5710
Lat. acc g's	0	.015	.022	.031	.073
Turning radius ft.					
5 kts	∞	148	101	72	30
15 kts	∞	1330	905	642	271
25 kts	∞	3700	2520	1790	760
Per cent installed LHP used for maneuver	0	28.6	44.5	54.5	100.

.04 g acceleration

.06 g deceleration.

6.4 FINAL STUDY CONFIGURATION

The considerations outlined in this chapter have led to the configuration shown in Figures VI-5 and VI-6 consisting of a minimum performance vehicle suited to the basic LOTS mission, utilizing modular power plants and interchangeable units as far as possible. Flush intakes to the lift system power plants are utilized to facilitate loading and unloading while alongside a ship or dock in heavy weather. It is considered that the low-speed and short range requirements for the LOTS carrier make more sophisticated intakes unnecessary. The complexity of an integrated lift-propulsion system cannot be justified without considerable design refinement, so a simple separate propulsion system is chosen in which the thrust may be oriented as desired for firm and additional control.

An attempt has been made to provide maneuver control with the aid of side-thrusting controllable louvers, but maneuvering can be conducted satisfactorily only at low speeds, or with the aid of the differentially controllable propulsion propellers. Hence, this vehicle is restricted to over-water missions in most cases.

To offset the limited maneuvering capability of the vehicle, ramps have been provided at both ends for loading and unloading purposes. Twin bow cabins accommodate a crew with dual controls to facilitate close hauled handling. Bow sections have been designed to absorb most of the shock of wave impact, and contain the necessary mooring gear. The stern sections contain auxiliary power units for water maneuverability.

The lift power modules, a typical section of which is shown in Figure VII-9, utilizes the simplest possible "straight-through" flow system with engine cooling taking place automatically. The choice of the "Gazelle Junior" engine is ideally suited since this engine can be mounted vertically.

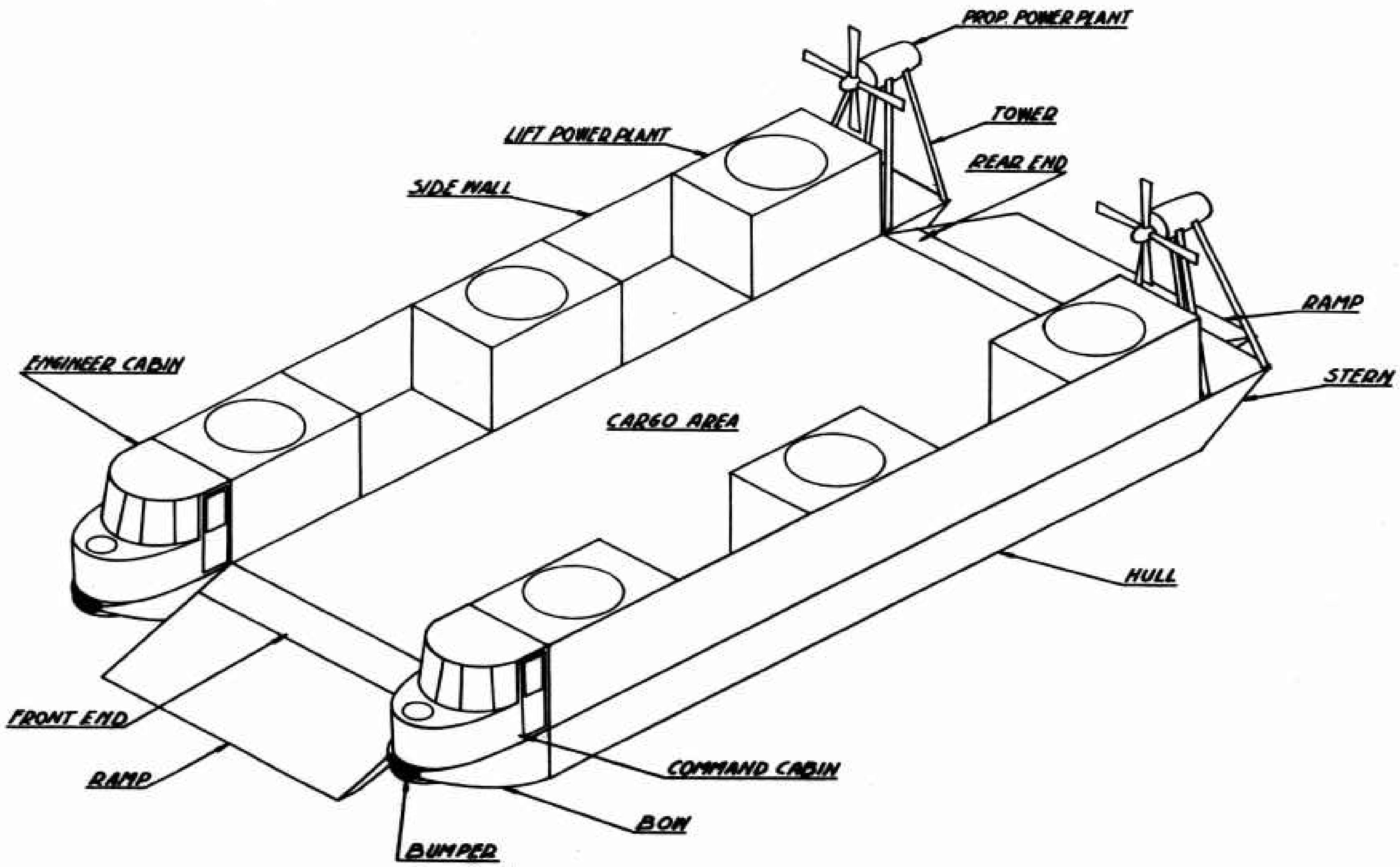


FIG VI-6 ISOMETRIC VIEW OF VEHICLE

6.5 PRELIMINARY WEIGHT AND BALANCE

In this chapter there are a number of weight items estimated in order to reach a gross weight figure for performance analysis. Utmost accuracy is not possible but the results are not harmed by a number of small errors in estimating; overs and unders will cancel each other.

Prior to preparing shear and bending moment data, it is necessary to refine the weight estimates and compute weight and balance and mass moment of inertia in accordance with the proposed design. Table VI-3 contains the data. The "final-preliminary" figures (this term is coined to indicate that the figures are as precise as they can be prior to structural proportioning and stress analysis) yield these characteristics:

Gross weight	-	75,372 pounds
c.g. location	-	Station 438
radius of gyration	-	171 inches.

6.6 SHEAR AND BENDING MOMENT CURVES

Shear and bending moments are in Figures VI-7, VI-8, and VI-9. The first of these curves contains the so-called one-g case which is not a complete shear or bending moment diagram due to the absence of any reactions. It is a useful device in preparing the diagrams for a variety of specified loading situations where the applied forces are reacted, in whole or in part, by inertia forces. Two cases are presented in detail: the one where a bow acceleration of 8g* is experienced (acceleration component computation is shown in paragraph 4.2)due to a force on the bow as would result for example in slamming, and the other where 6g translational acceleration is experienced due to two end forces as would result from a hard landing with gear contacting the ground. Pertinent data on force location is contained on the curves. As would be expected the latter is far more severe. If due to some special cargo requirement, load distribution were to be radically changed and large masses placed near the extremities, this result would not be applicable. In such an event, moments due to translational accelerations would be increased for the bow slamming case.

*To employ a 1.5 safety factor the peak values off the curves are simply scaled up in direct proportion.

TABLE VI-3
PRELIMINARY WEIGHT AND BALANCE DATA

	Item	Weight (lb.)	Station (in.)	Moment (in.-lb.)	Moment of Inertia (in.^{2} - lb.)
1.	Cabin (C)	1010	68.35	69,041	4,718,952
2.	Bow (P)	300	66	19,800	1,302,000
3.	Cabin (E)	1010	68.35	69,041	4,718,952
4.	Bow (S)	300	66	19,800	1,302,000
5.	Front End	625	92	57,500	5,290,000
6.	Ramp	600	66	39,600	2,608,000
7.	Main Struct. Mod -1	1616	141	227,856	32,127,695
8.	-2	1616	207	334,512	69,243,984
9.	-3	1616	273	441,168	120,438,864
10.	-4	1616	339	547,824	185,712,336
11.	-5	1616	405	654,480	265,064,400
12.	-6	1616	471	761,136	358,495,056
13.	-7	1616	537	867,792	466,004,304
14.	-8	1616	603	974,443	587,592,144
15.	-9	1616	669	1,081,104	723,258,576
16.	-10	1616	735	1,187,760	873,003,600
17.	Wall -1	41	306	12,546	3,839,076
18.	-2	41	306	12,546	3,839,076
19.	-3	41	570	23,370	13,320,900
20.	-4	41	570	23,370	13,320,900
21.	L. Power Plant Mod. -1	1848	174	321,552	55,950,040
22.	-2	1848	174	321,552	55,950,048
23.	-3	1848	384	709,632	272,498,688
24.	-4	1848	384	709,632	272,498,688
25.	-5	1848	648	1,197,504	775,982,592
26.	-6	1848	648	1,197,504	775,982,592

TABLE VI-3 (continued)
PRELIMINARY WEIGHT AND BALANCE DATA

Item	Weight (lb.)	Station (in.)	Moment (in. - lb.)	Moment of Inertia ($in.^2$ - lb.)
27. Rear End	625	784	57,500	5,290,000
28. Stern (P)	305	791.3	241,347	190,977,881
29. Stern (S)	305	791.3	241,347	190,977,881
30. Tower (P)	120	803	96,360	77,377,080
31. Tower (S)	120	803	96,360	77,377,080
32. Prop. Power Plant Mod. -P	905	798	722,190	576,307,620
33. Prop. Power Plant Mod. -S	905	798	722,190	576,307,620
34. Prop. Fuel Tank -P	130	810	105,300	85,293,000
35. Prop. Fuel Tank -S	130	810	105,300	85,293,000
36. Ramp	600	810	486,000	393,660,000
37. Cargo Floor -1	239	141	33,699	4,751,559
38. -2	239	207	49,473	10,240,911
39. -3	239	273	65,247	17,812,431
40. -4	239	339	81,721	27,466,119
41. -5	239	405	96,795	39,201,975
42. -6	239	471	112,569	53,019,999
43. -7	239	537	128,343	68,920,191
44. -8	239	603	144,117	86,902,551
45. -9	239	669	159,891	106,967,079
46. -10	239	735	175,665	129,113,775
47. Crewman -1	340	72	24,480	1,762,560
48. -2	340	72	24,480	1,762,560
49. Fuel -1	450	129	58,050	7,488,450
50. -2	450	129	58,050	7,488,450

TABLE VI-3 (continued)
PRELIMINARY WEIGHT AND BALANCE DATA

Item	Weight (lb.)	Station (in.)	Moment (in. - lb.)	Moment of Inertia ($in.^2$ - lb.)
51. Fuel -3	450	219	98,550	21,582,450
52. -4	450	219	98,550	21,582,450
53. -5	450	393	176,850	69,502,050
54. -6	450	393	176,850	69,502,050
55. -7	450	483	217,350	104,980,050
56. -8	450	483	217,350	104,980,050
57. -9	450	657	295,650	194,242,050
58. -10	450	657	295,650	194,242,050
59. -11	450	747	336,150	251,104,050
60. -12	450	747	336,150	251,104,050
61. -13	750	810	607,500	492,075,000
62. -14	750	810	607,500	492,075,000
63. Cargo	30,000	438.5	13,155,000	5,818,867,500
Total	75,372			16,452,790,235

c.g. Station 438

Radius of gyration 171.36 inches.

Note: Load lumped at 6 stations equally spaced from 378.5 to 498.5 for Moment of Inertia compuration.

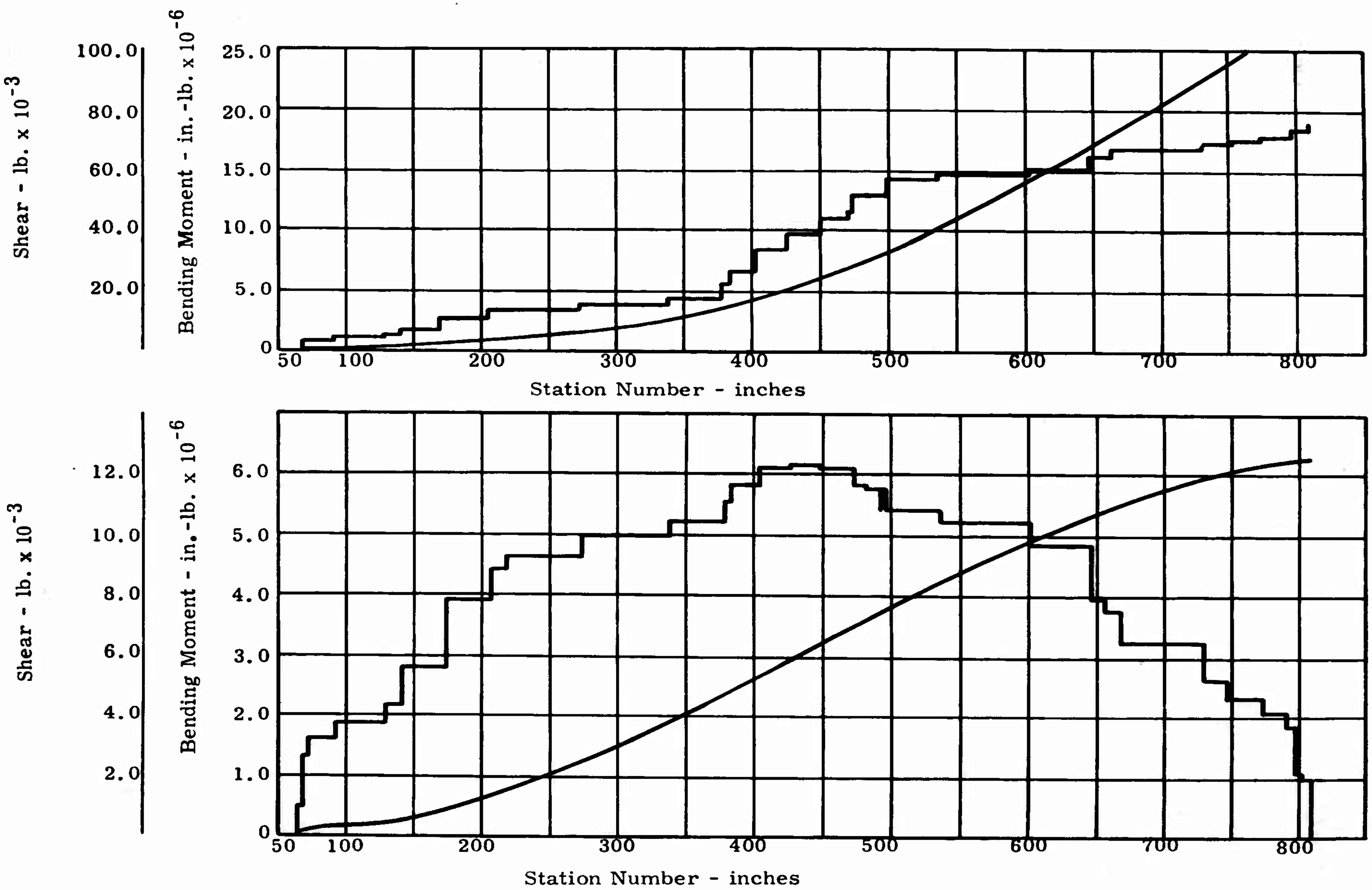


Fig. VI-7. Unit Shear And Bending Moment Curves

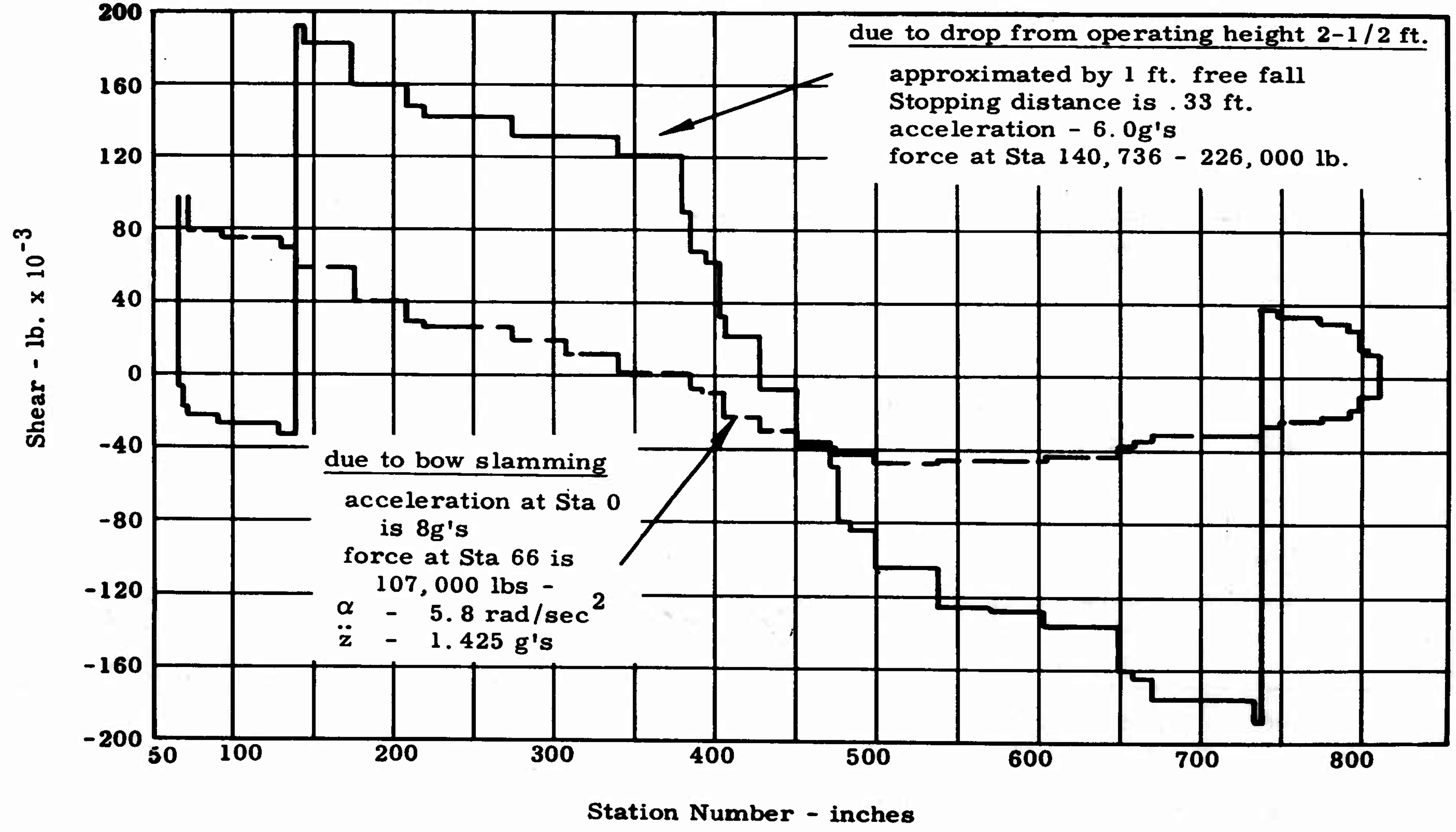


Fig. VI-8. Shear Load For Two Loading Conditions

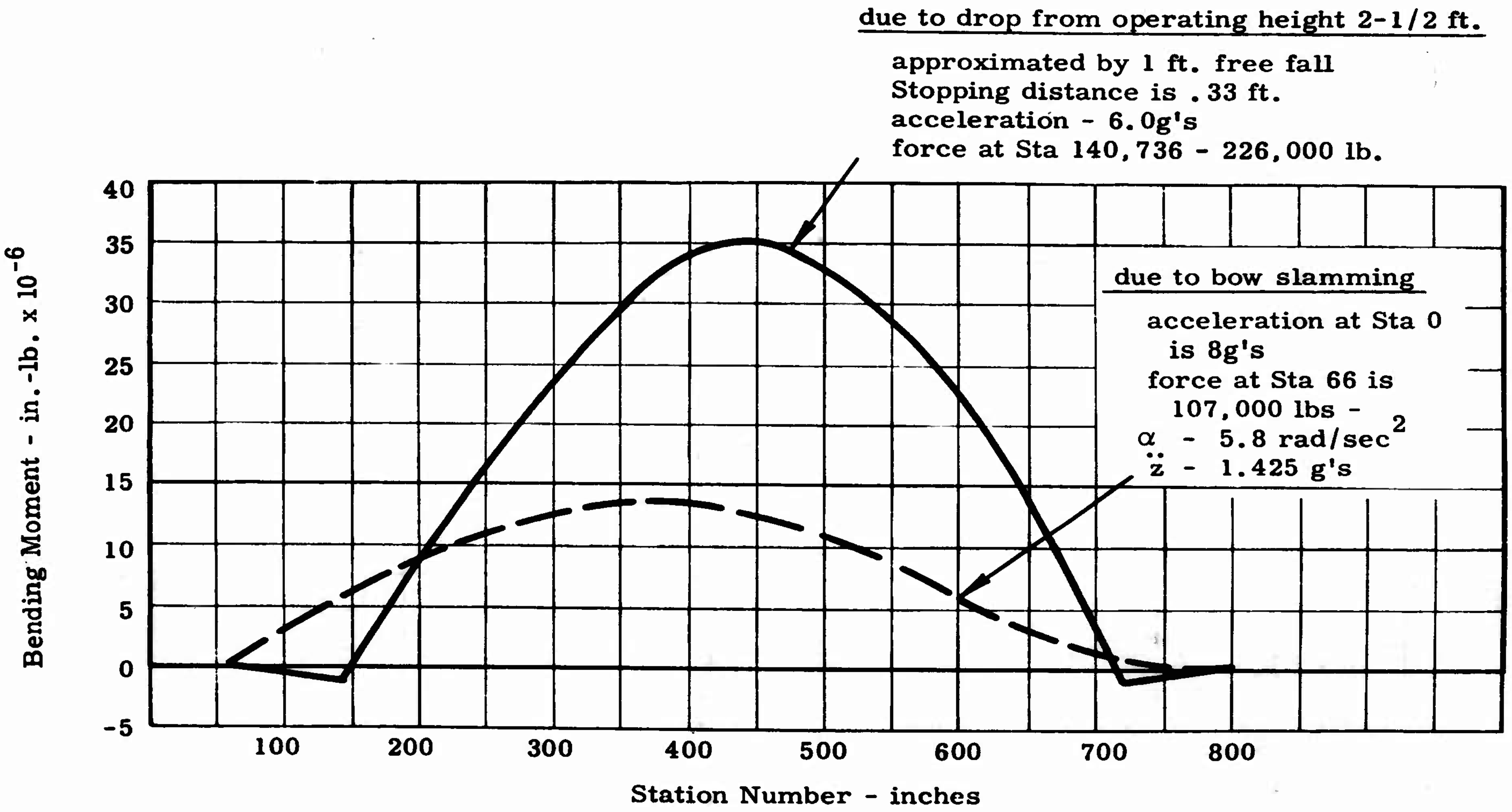


Fig. VI-9. Bending Moment For Two Loading Conditions

One other characteristic in the curves may be noticed. The ratio of shear to bending moment is higher than encountered in most aeronautical applications. This is due to the proportions of the vehicle which result in a fairly stubby body and also to load distributions. The consequence of this is apparent in the stress computation where the shear influence is more pronounced.

CHAPTER VII

THE DESIGN OF A LOTS CARRIER GEM FOR MAXIMUM TRANSPORTABILITY

7.1 BASIC CONSIDERATIONS

At this point, the question to be considered is: What sort of breakdown will achieve transportability as an additional requirement superposed on all the others? There are numerous subsidiary questions to be answered in the process; they are:

- Do these analytical investigations indicate technical feasibility?

- What are the practical problems to be overcome in gaining the advantage of maximum transportability?

- Can sectionalization be applied to existing designs or must a design be inherently suitable?

- Which is the preferred approach from among sectionalization, modular breakdown, collapsibility, inflatability?

- What are the penalties associated with transportability, e.g., additional structural weight, higher cost?

7.1.2 Definitions

7.1.2.1 Module is the term applied to a subdivision when a number of similar units can be combined into a working assembly; obviously, for a GEM several different kinds of modules will be required.

7.1.2.2 Section is the term applied to a subdivision when the whole is broken into parts based on physical location. In this case the prospect for a number of identical subdivisions is less. The degree to which sections and modules differ is influenced by the nature of the totality. In the case of a rectangular planform and nearly homogeneous distribution there might be a practical merging of the two.

7.1.2.3 A collapsible unit is one which is articulated so as to occupy less space in its inactive configuration. Examples are readily available in the case of the wings of carrier-based aircraft and in domestic folding tables.

7.2 SECTIONALIZED BREAKDOWN

Given a vehicle design, this type of a breakdown is the direct way to accomplish transportability. If it were necessary to limit time in the design process this would be mandatory; for smaller vehicles, just outside transportability limits, the designer would probably not justify an attempt to optimize but would proceed on the basis of transverse or longitudinal sections.

In attempting to reach optimal design, the considerations listed below would need be taken into account (applicable to vehicles from 25 tons gross weight upward).

7.2.1 Favorable to Sectionalization

7.2.1.1 Fewer parts. Providing that the very outside limits of transportability may be approached, a vehicle will have the fewest joints. This tends to lower assembly time, and leads to the following points.

7.2.1.2 Fewer transfer joints. This is a definite weight saving for joints can be a considerable part of the structural weight and obviously the fewer joints the least penalty. (See Figure X-1 for numerical indications.)

7.2.1.3 Fewer control interconnects. Fewer breaks in control links facilitates assembly and rigging and decreases probability of maloperation.

7.2.1.4 Applicable to existing designs. The transportability of existing designs can be improved by sectionalizing even when the feature was not incorporated at the inception

of the design. In most instances the logical location for sectionalizing can be determined by inspection of the various autonomous components.

7.2.2 Unfavorable to Sectionalization

7.2.2.1 Handling difficulties. Loading, unloading, and assembly would require more elaborate facilities and greater load capacity in lifting equipment.

7.2.2.2 Irregular forms. The forms could not be expected to be prismatical with square corners unless a deliberate effort is made in that direction. In the case of an existing design being sectionalized, the probable consequences are odd forms, complicated crating and dunnage, and waste of transport space.

7.2.2.3 Limited transportability. Some sections could present difficulties in attainment of the dimensions consistent with maximum transportability.

An application of sectionalization, in brief form, is seen in Figures VII-1 and VII-2, and Table VII-1. The approximate dimensions of this proposed design for a cargo lighter are 70 feet by 38 feet by 21 feet high and its gross weight is 37-1/2 tons. (This is in the same bracket as the vehicle on which the study is based.) It may be seen that the total volumes are very high even sectioning into the 17 and 23 parts. Furthermore, and of most significance, is the fact that transportability is still very poor. In Case I, 35 per cent of the sections, and in Case II, 10 per cent of the sections, cannot be accommodated in the largest current transport aircraft, the C-133.

7.3 MODULAR BREAKDOWN

This kind of vehicle breakdown can only be applied when the design has anticipated the requirement, such as the case of the LOTS carrier outlined in the previous section. Utmost simplicity has prevailed:

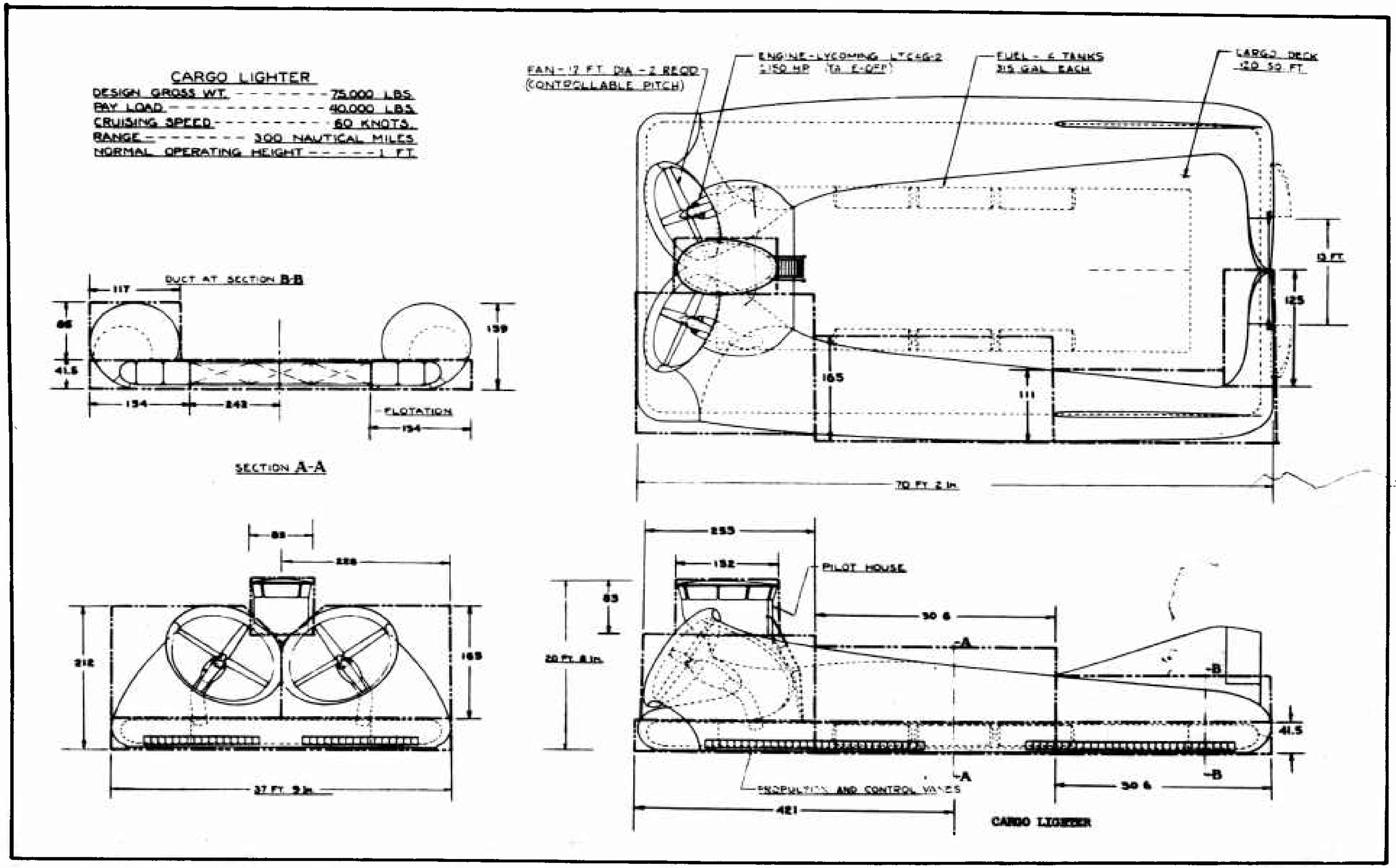


Fig. VII-I Sectionalization of an Existing Design-Case I

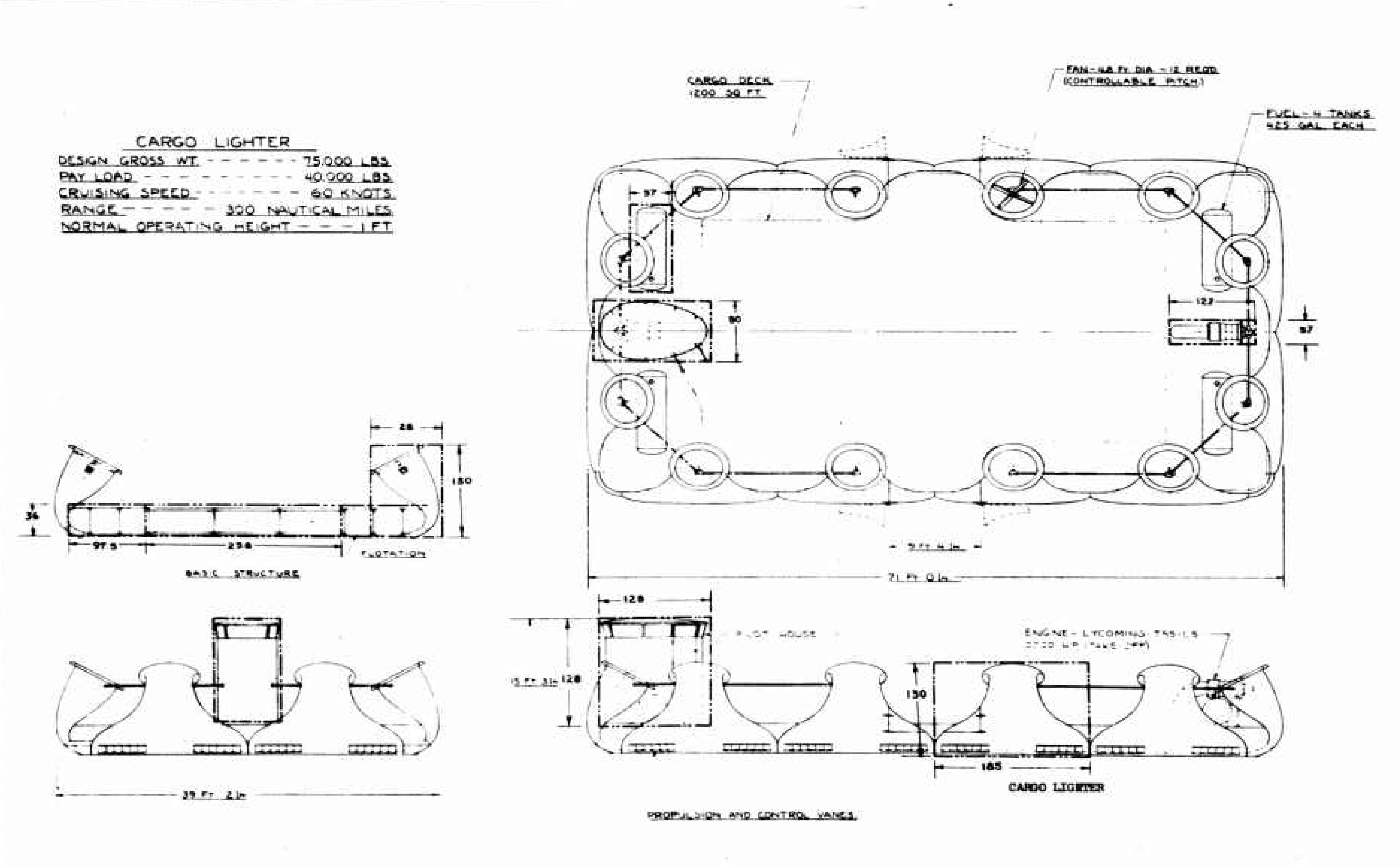


Fig. VII-2 Sectionalization of An Existing Design-Case II

TABLE VII-1
TYPICAL CARGO LIGHTER (GEM) SECTIONALIZATION

Case I

Part	No. of Part	Length (ft.)	Width (ft.)	Height (ft.)	Vol. ea. Cu. ft.	Vol. Total (ft.)	Air Transp.
1	2	19.6	19	13.8	5130	10260	No
2	1	11	6.9	6.9	523	1046	
3	2	25.5	13.8	9.3	3272	6544	No
4	2	25.5	9.3	5.8	1375	2750	
5	4	35.1	11.2	3.5	1353	5412	
6	2	35.1	20.2	3.5	2481	4962	No
7	2	16.2	6.3	4.6	469	938	
8	2	18.9	9.2	0.7	12	242	
Total	17					32,154	

Case II

Part	No. of Part	Length (ft.)	Width (ft.)	Height (ft.)	Vol. ea. Cu. ft.	Vol. Total (ft.)	Air Transp.
1	12	15.4	7	10.8	1170	14040	
2	2	10.2	3.1	3.1	95	190	
3	2	10.2	3.1	3.1	95	190	
4	1	10.7	6.7	10.7	795	795	
5	4	35.5	8.2	2.9	845	3380	
6	2	35.5	19.9	2.9	2045	4090	No
Total	23					22,685	

Note: Refer to Figures VII-1, VII-2 for applicable vehicles.

air passages have no unnecessary duct work, the planform is uncluttered, power for the lifting system is symmetrically disposed in units of a convenient size, needless complication of lift and propulsion by interconnection of power sources is avoided. However, there is no appreciable sacrifice in the characteristics which must be incorporated in the vehicle. It may be seen in the detailed design work and reference to Table II-1 that structure weight (at 40.3 per cent of gross weight) is not out of line with the current state-of-the-art; i.e., other GEMs of rugged design also have a structure weight of 40 per cent of gross weight.

When a number of similar modules can be put together into a working vehicle there are a number of advantages:

- Size of modules can be consistent with maximum transportability.
- Modules and their joints can be designed for expedited assembly.
- Manufacture of a large quantity of identical units will tend to lower cost (although precision requirements for good fitting will make any net gain very small).
- Several different vehicle sizes can be put together from standard modules.

On the other hand:

- Modular construction does not minimize the number of transfer joints and consequently there is some extra weight penalty.
- Semi-precision fit is implied and detracts from the maintenance advantage otherwise expected.

7.4 VEHICLE DESIGN FOR MODULAR BREAKDOWN

7.4.1 General Breakdown

Referring to Figure VI-2 for the general vehicle arrangement, an evaluation of all possible breakdowns resulted in the following preferred solution:

Main structural modules (eight identical interior units, two end units)	10 units
Power plant (lift) modules	6 units
Propulsion units	2 units
Front and rear ends	2 units
Bow	2 units
Stern	2 units
Cabin	2 units
Ramps	2 units

See Figure VII-3 for this breakdown. The main modules are transversally disposed units in order that the number of identical units be maximized. Longitudinal units only would have limited transportability (the C-130 will not accommodate the length). The 10 units as designed are on the order of 1,600 pounds which is on the high side for articles which might need to be manhandled in loading, unloading, and assembly operations. A compromise is necessary, of course, since lowering the weight per module means more joints. With adequate mechanized handling equipment and transport aircraft in the C-133 category, the main modules could be handled in pairs resulting in only five main structural loads.

An approach to a more favorable joint array would be an increase in length-to-beam ratio. As it now stands, the design has a cargo space, long enough to accommodate many useful combinations of military

LIST OF MODULES

ITEM	DESCRIPTION	No. OF MOD.
1	RAMP	2
2	END BODY	2
3	BOW	2
4	CABIN	2
5	FRONT & REAR MAIN STRUCTURE	2
6	INTERIOR MAIN STRUCTURE	8
7	LIFT POWER PLANT	6
8	SIDE PANEL	4
9	COVER PLATE	4
10	STERN	2
11	PROPULSION POWER PLANT	2
12	SUPPORTING TOWER	2
13	PROPELLER	2
	TOTAL	40

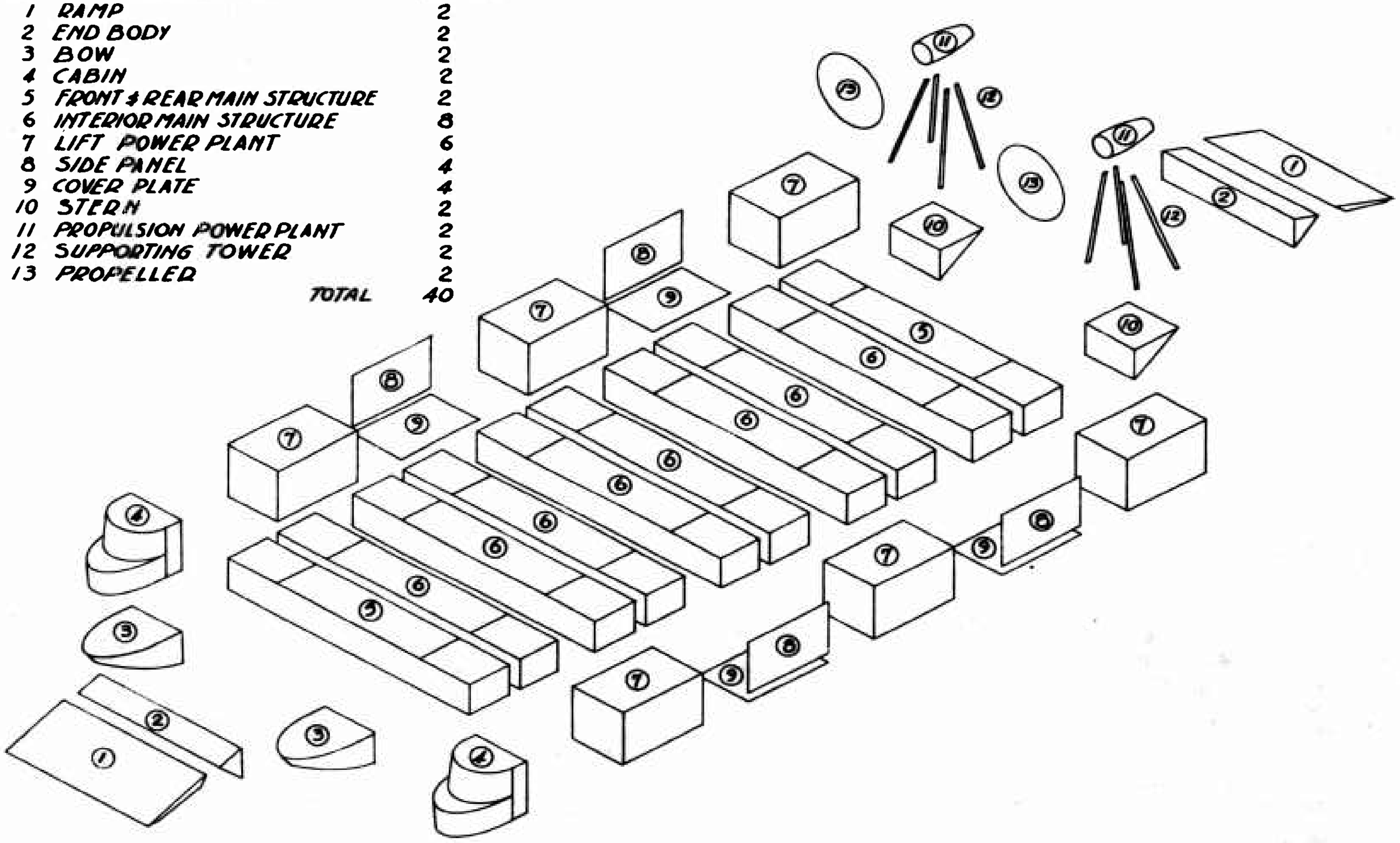


FIG VII-3 MODULAR BREAKDOWN FOR L.O.T.S CARRIER

vehicles, weapons, and miscellaneous cargo. A further lengthening detracts from the efficiency of the lift system which is optimized by lessening the cushion periphery for a given area.

7.4.2 Main Structural Modules

The main modules, when joined together, form the largest part of the structure. Functions performed are as follows:

- Primary structure
- Flotation
- Cargo platform
- Lift and stability distribution systems (jets)
- Control jets (side)
- Supporting structure for the following:
  - Cabins and bows
  - Front and rear ends
  - Sterns
  - Lift power plant modules
  - Propulsion power plant modules

The report contains many references to Main Structural Modules frequently without identifying the variations which occur. Eight of the ten modules are identical in structural members, attaching provisions, watertight volumes, and side duct sections -- these are the <u>interior main structural modules</u>. The modules going into the two end positions are similar structurally and in external dimensions but contain additional ducting in order to complete the peripheral annular-jet -- these are the <u>end main structural modules</u>. See Figures VII-4 and VII-5.

<u>Dimensions</u>

Length	408 inches = 34 feet
Width	66 inches = 5.5 feet
Height	48 inches = 4 feet

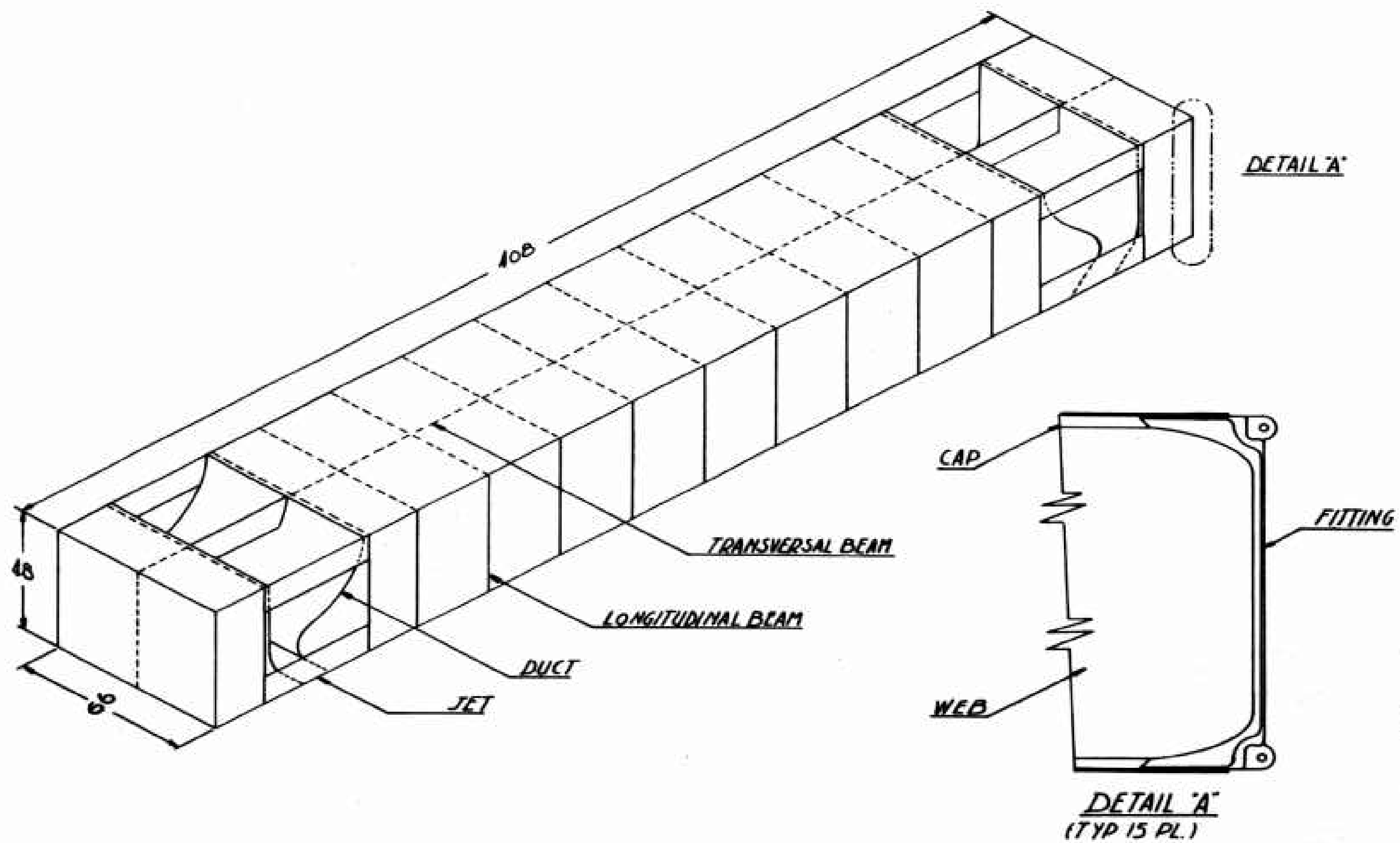


FIG VII-4 MAIN STRUCTURAL MODULE

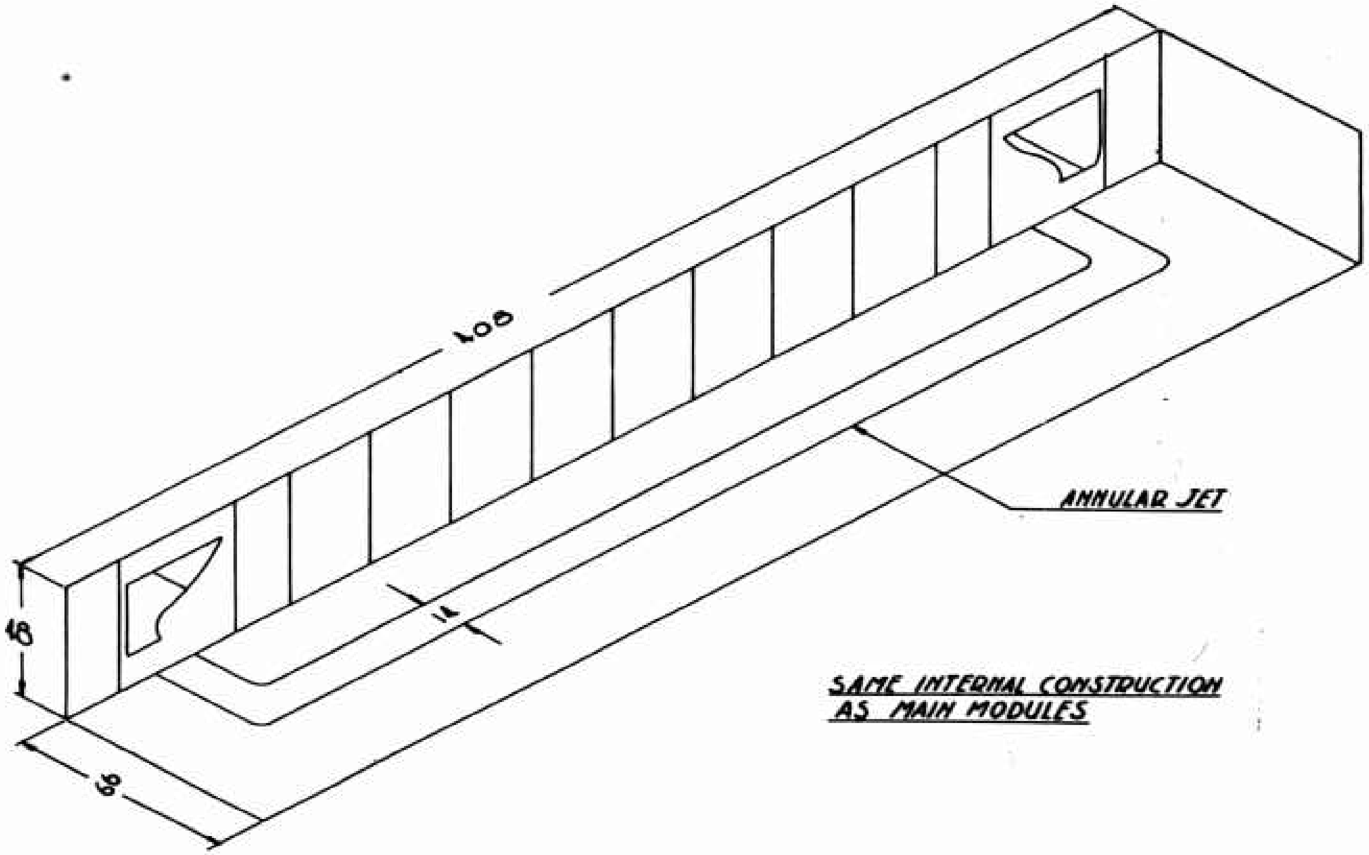


FIG VII-5 MAIN STRUCTURAL VARIATION FOR FORE AND AFT POSITIONS

Weight

$$W = 1{,}616 \text{ pounds}.$$

The modules are of aluminum alloy based on previously detailed discussions and contain 15 longitudinal beams, each having load-carrying transfer joints, and 3 transverse beams to carry the lateral bending moments and shear loads. This network of beams together with the bottom and top skin provide 16 watertight compartments in the center section under the cargo floor providing a volume of 27.5 cubic feet each. The total combined buoyancy volume of the craft is thus equal to 3,960 cubic feet envisioning that a weight of 243,204 pounds can be supported.

Under normal loading conditions, the displacement equals 75,000 pounds and under overload conditions 135,000 pounds. The excess of buoyancy is therefore:

$$\frac{243{,}204}{135{,}000} - 1 = 1.8 - 1 = 0.8.$$

The draft under normal load conditions is

$$\text{Draft} = \frac{75{,}000}{62.36 \times 88.0} = 1.36 \text{ feet}$$

and under overload conditions

$$\text{Draft (overload)} = \frac{135{,}000}{62.36 \times 880} = 2.46 \text{ feet}.$$

Due to the annular jet configuration of the GEM it is necessary to incorporate in the end modules the front and rear lift jets. When assembled, the final result provides a cargo platform of 1,100 square feet and a cargo floor designed for a cargo density of 300 pounds per square foot.

7.4.3 Power Plant Module

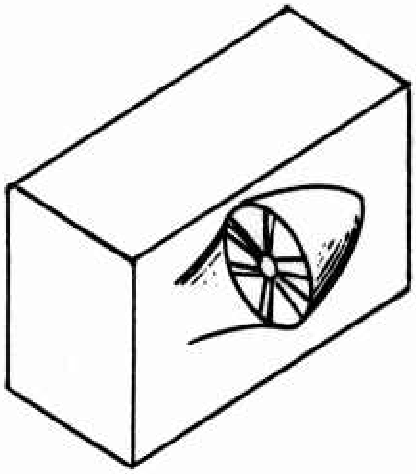
In addition to the module structure, the problem of engine selection is also included here as the engine itself has so much influence on the module. Compromise is evident again as the power plant module configuration is evolved. Consideration of the following is a vital step in the design:

- Fan and air ducting space requirements
- Air inlet geometry
- Engine size and weight
- Limits of transportability to include over-all dimensions and weight for efficient handling and assembly.

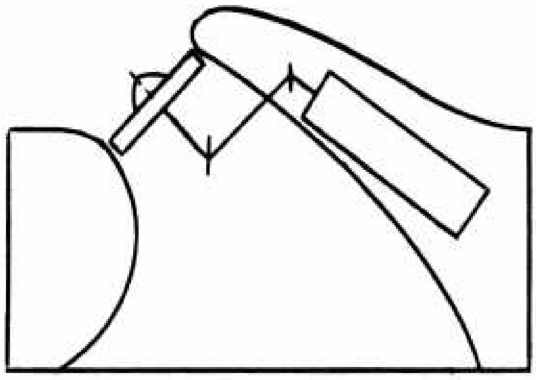
7.4.3.1 Fan Size

Fan size (see Chapter V, Cushion Pressure, etc.,), for optimum handling of the cushion air flow results in a fan 5.5 feet in diameter (for each of six units). The predisposition to six units permits a start to be made and can be likened to the iteration which appeared in the general design problem of the vehicle.

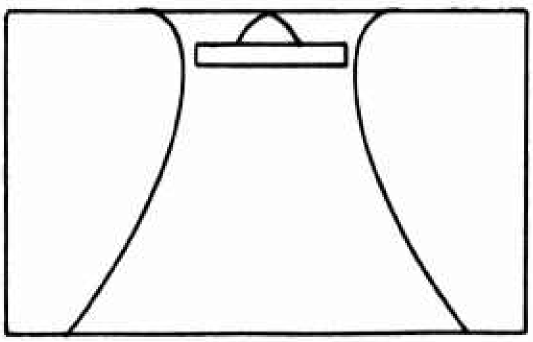
Effectiveness of an air inlet design is a detailed aerodynamic problem which will be considered only briefly. For a vehicle operating in the low speed regime (as does the LOTS carrier), the problem is much less important than in the case of vehicles where ram efficiency and over-all drag are critical. For the LOTS carrier, minimum weight will pay off much more quickly. A side air intake dictates right-hand and left-hand modules not interchangeable and offers least chance of minimizing ducting. Intake of air from the top of the module will result in a straight line arrangement for inlet, fan, reduction gearing, and engine. An air scoop is not considered essential and would, in fact, be vulnerable to frequent damage during cargo handling. A canted inlet offers an advantage in space utilization (see Figures VII-6, VII-7) but an arrangement as shown would complicate engine removal for maintenance purposes. The preferred arrangement -- flush, top intake -- is included in the figures.



Side Intake

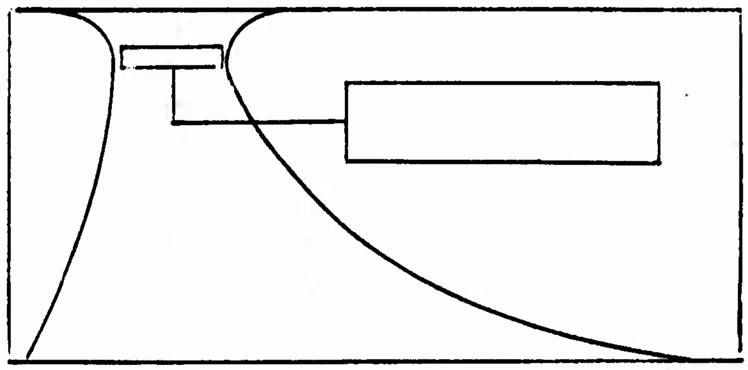


Top Canted Intake

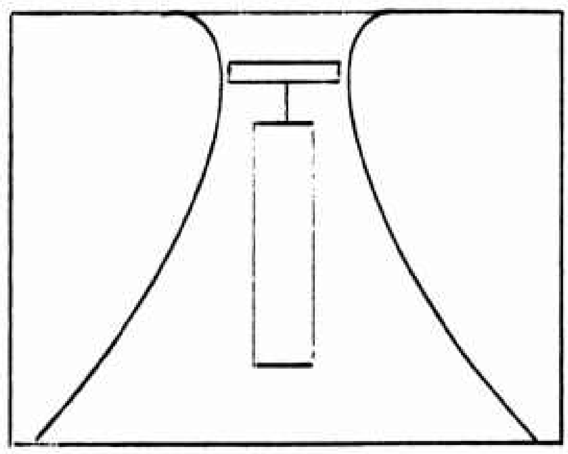


Top Flush Intake

Fig. VII-6. Air Intake Variations



Engine Horizontal Installation



Engine Vertical Installation

Fig. VII-7. Engine Position Variations

7.4.3.2 Engine Selection

The selection of an engine was studied in detail, as this influences the module design to such a large degree. The total installed power of the lift system must be 5,400 SHP (using the rating for one-hour operation). The engines which are in the range of suitability are tabulated in Table VII-2. Larger engines (not included in the table) produce a module which is not transportable. The engines which could be used in a four-engine arrangement result in excess capacity and weight and a module with longer air flow paths. Among the engines which are suited to a six-engine arrangement, the Napier Gazelle Junior is the only one which turns at a speed not requiring any additional gear reduction. The SHP/W column of the data table is thus not indicative of power plant installed weight. On a SFC comparison the G.E.T-58-GE-6 shows up well but its extra length detracts from its utility. The Lycoming engines are not as good a power match as the others. For this application the Gazelle Junior is selected. For a longer range vehicle a more detailed analysis would be required in order to determine where the SFC disadvantage overrides the other advantages. The selected engine has been designed primarily for helicopter application and may be mounted vertically, horizontally, or in any intermediate position as required. The engine is a complete power package with a self-contained oil system and requires external connections only for the fuel lines and cockpit controls making it a particularly good choice for a modular application. Pertinent data are as follows:

7.4.3.2.1 Engine Dimensions

Maximum diameter	33 inches
Length including accessories	54 inches
Engine weight (less starter, generator)	495 pounds

7.4.3.3 Fuel Requirements for Lift

Range	50 miles
Speed	40 miles per hour
Power required	5,400 horsepower

TABLE VII-2
SELECTION OF ENGINE CHARACTERISTICS OF ENGINES CONSIDERED

Model	Power										
	Maximum		Rated				Dimensions				
	SHP	RPM_o	RPM_o	SHP	RPM_o	SFC	L	Ø	W	SHP/W	
G. E.											
T-58-GE-6	1050	26,300	6,000	900	6000	0.64	72.2	16	348	3.2	(1) (6)
T-58-GE-8	1250	25,600	6,000	1050	6000	0.64	72.2	16	366	3.42	(1) (6)
T-64-GE-2	2850	13,600	5,200	2270	5200	0.522	91	25	864	3.1	(2) (6)
T-64-GE-6	2850	13,600	13,600	2270	13600	0.522	83	24	713	3.7	(2) (5)
Lycoming											
T-53-L-5	960	25,240	6,607	825	6610	.719	47.61	23.0	485	2.07	(6)
T-53-L-9	1100	26,000	6,610	900	6610	.702	47.61	23.0	485	2.37	(6)
T-55-L-3	1900	18,660	6,750	1700	6460	.690	44.04	24.25	600	3.17	(6)
T-55-L-5	2800	18,660	14,550	1850	14000	.660	44.04	24.25	570	3.86	(5)
Napier											
Gazelle N Ga. 2	1650	20,400	3,000	1300[3] 1050[4]	3000	0.769	70.0	33.5	830	1.99	(2)
Gazelle N Ga. 4	2000	20,400		1575[3] 1350[4]	3000	0.705	70.0	33.5	900	2.22	(2)
Gazelle Junior[1]	1070			900[3] 800[4]	3000		54.0	33.0	495	1.86	(2)

SHP Shaft Horsepower
RPM_g RPM gas generator
RPM_o RPM output shaft

(1) To be mounted at not more than 30° from horizontal.
(2) To be mounted on any position either vertical or horizontal.
(3) Maximum 1 hour rating.
(4) Maximum continuous rating.
(5) Do not include reduction gear.
(6) Further reduction of RPM required.

Fuel: 0.80 lb/hp-hr x $5,400 \times \frac{50}{40} = 5,400$ pounds

(reserve included due to reduced power running empty on return leg)

Volume of tanks	5,400/62.35 x 0.8 = 108 cubic feet (total all modules)
Capacity of tanks	108 x 7.48 = 808 gallons

7.4.3.4 <u>Engine Supports</u>

Four arms to pick up mounting pads, radially disposed, are consistent with the engine layout as the torque reaction is through a beam attached to the gear box.

7.4.3.5 <u>Fuel Tanks</u>

Fuel tanks appear to be no problem in the module design. Referring to Figure VII-8, it is apparent that space for integral tanks is adequate. The total volume for the six modules is estimated to be 225 cubic feet, ensuring a total capacity of 1692 gallons.

The over-all picture of the power plant module is given by Figure VII-9, the major characteristic being:

- length — 132 inches (11 feet)
- width — 84 inches (7 feet)
- height — 84 inches (7 feet)
- weight, dry — 1,850 pounds

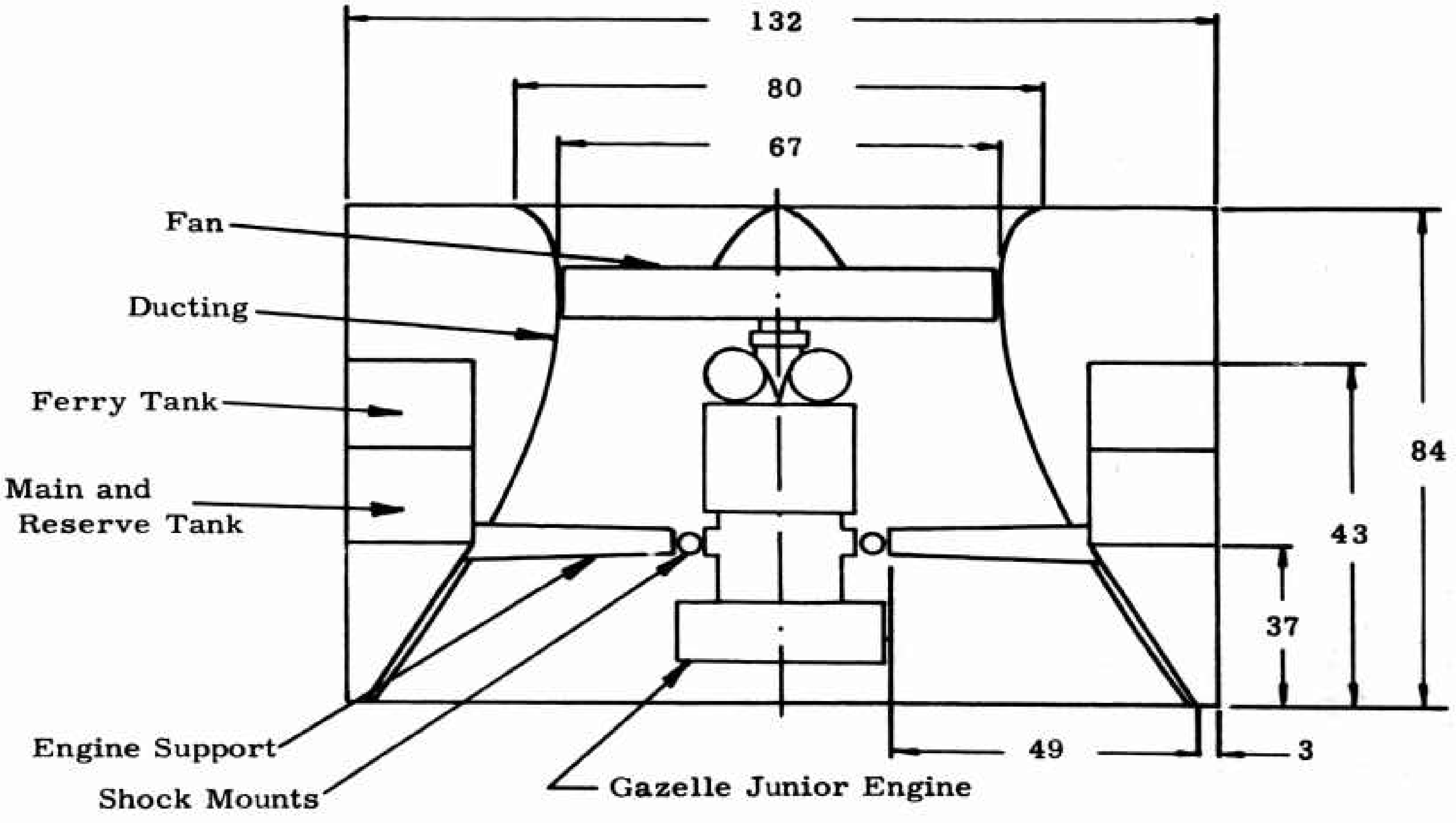


Fig. III-9. Lift Power Plant Module Cross-Section

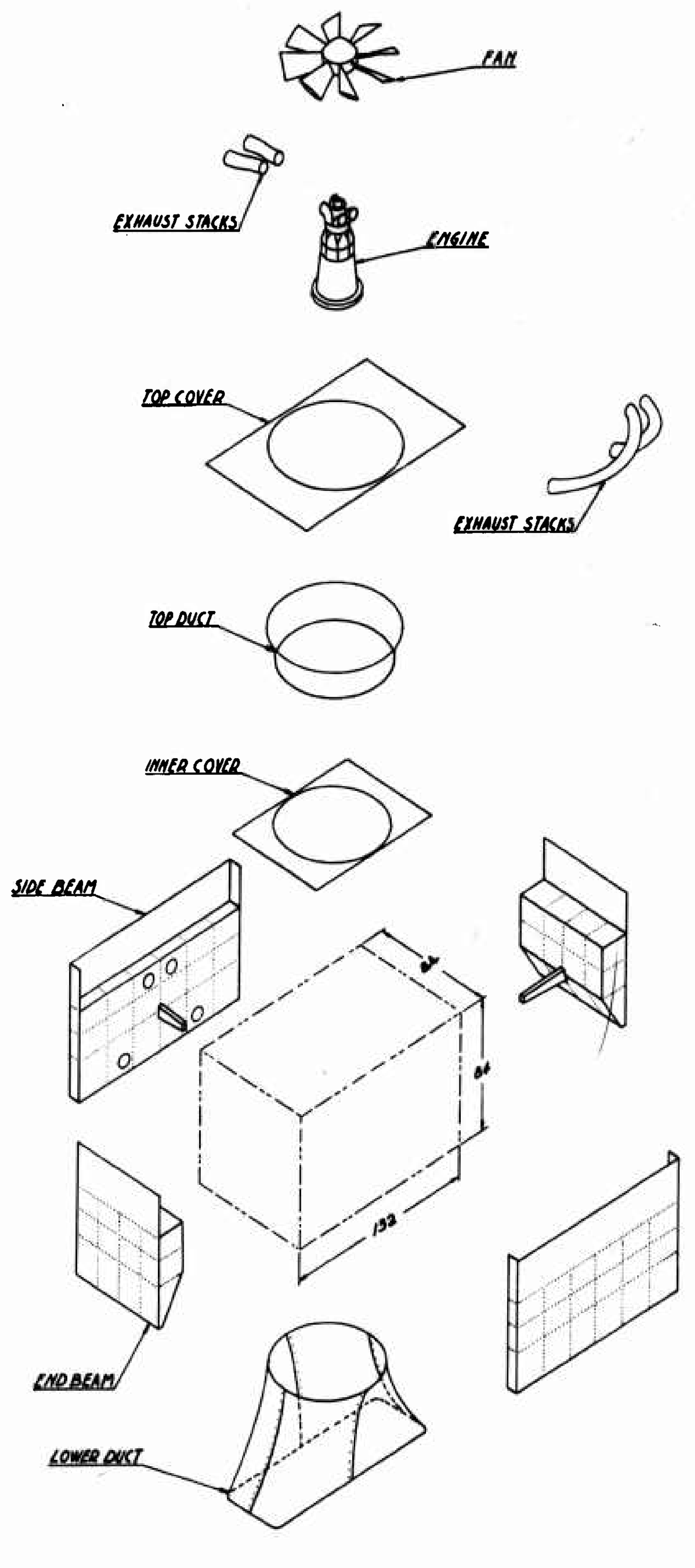


FIG VII-9 POWER PLANT MODULE EXPLODED VIEW

The dimensions are all within transportability requirements, the form lending itself to shipping and handling by its regular shape and simplicity. No special tools are required to assemble the unit into the LOTS craft. Should mechanized equipment be unavailable for handling the unit, manpower could be substituted in a similar manner to which pontoons are handled in military bridge assembly.

7.4.4 Cabin Modules

Due to the LOTS carrier configuration which makes a central ramp advisable, the craft is fitted with two cabins of the same structural configuration. The left-hand unit is adapted to command, communications, and navigation; the right-hand unit is adapted to engine control. These modules (see Figure VII-10) are supported by the bow modules of the craft and are completely self-contained units, having external connections as follows:

- Structural attachments
- Controls electrical interconnection
- Communications interconnects.

Both cabins have doors on either side for communicating with the cargo deck or for boarding the craft from outside. The height and width dimensions match the dimensions of the lift power modules (7 x 7 feet), the length being 9 feet. A bow hatch on each cabin for mooring operations is included. The empty weight of both cabins is approximately the same (approximately 1,000 pounds).

7.4.4.1 Command Cabin

The command cabin (starboard side) has windows all around (three sides) and accommodations for two crew members -- the coxswain and the helmsman. As is indicated by the name, this cabin has all the necessary controls for governing the craft and some engine controls for emergency (override mode) operation.

The following controls, instruments and furnishings are incorporated in this cabin:

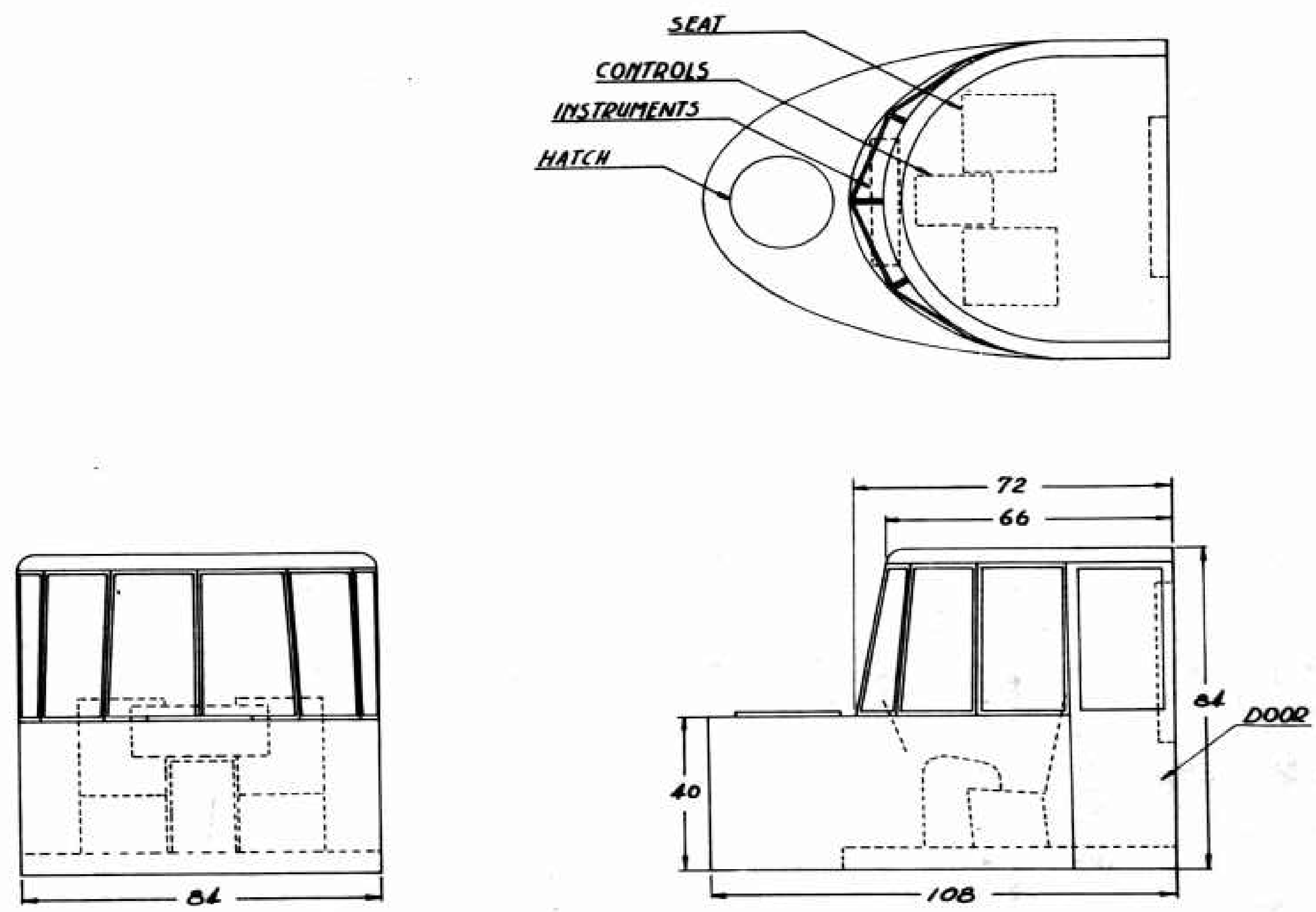


FIG VII-10 CABIN LAYOUT

Controls

- Lift (override)
- Side jets
- Propulsion (override)
- Steering
- Ramps operating system

Instruments

- Magnetic compass
- Directional gyro
- Radio compass
- Clock
- Speed indicator

Electronic

- Radio communications
- Radio navigation
- Intercommunication system

Furnishings

- Seats (2)
- Heat and vent system
- Fire extinguishers
- Lights and electrical system
- First-aid kit
- Very pistol
- Machete
- Bilge pump
- Data case
- Horn
- Bell
- Life preservers
- Batteries - emergency

7.4.4.2 Engineer's Cabin

The engine control cabin (starboard side) has the same construction but has less windows and differs from the command cabin in the

interior arrangement and furnishings. Its primary function is to provide accommodations for the machinist's mate and his assistant and will house all the engine controls and instruments, plus the electrical distribution system.

The following controls, instruments and furnishings are incorporated in this cabin:

Control

- Engine power lever
- Starting system
- Electrical distribution system
- Engine fire extinguisher system
- Heat and vent system

Instruments

- Engine instruments
- Electric generating system instruments

Furnishings

- Seats (2)
- Heat and vent system
- Fire extinguishers
- Boat hook
- Raft
- Tool kit
- Life preservers
- Anchor and winches
- Data case
- Bilge pump

7.4.5 Front and Rear Ends

The end modules may be seen in Figure VII-11. Their function is to provide sloping surfaces to alleviate water loads. Their construction is similar to the main modules and thereby additional buoyancy (not included in the quoted figures) is provided. Ramps are supported by these modules. Additional functions are provided by these modules. The front end facilitates passage of crew members from the deck to the cabins, and the rear end provides access to the propulsion power plants.

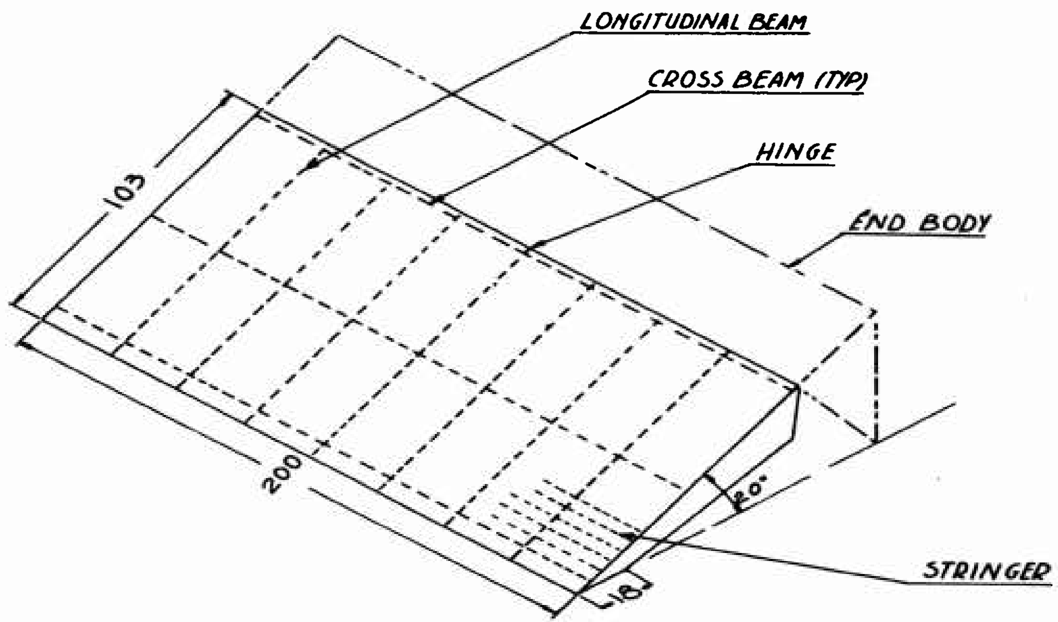


FIG VII-11 RAMP STRUCTURE AND INSTALLATION

7.4.6 Bow

It has been necessary to detach this portion of the structure from the cabin due to transportability considerations. The form of these modules follows closely the bow lines used in seaplane hull designs to provide for a reduction of spray, lower water resistance and minimum impact water pressure loads.

The type of construction used is consistent with seaplane construction practice and with the type used throughout the rest of the craft. This structure provides support to the cabins and is attached to the front of the main structure at both sides of the front end.

7.4.7 Ramps

Two ramps are provided; one is considered to be essential and the second facilitates unloading in the case where fork lift equipment is employed. Another advantage accrues from two ramps in that "roll-on/roll-off" operation is possible should the LOTS carrier be employed in ferry operations. See Figure VII-11 for the installation location and ramp details.

7.4.8 Stern

To provide for support of the propulsion power plants and their related equipment, stern sections attach to the main structure. They are similar to the front and rear end in construction and take the form shown in Figure VII-3.

7.4.9 Propulsion Power Plant Module

This module incorporates a T-53-L-5 Lycoming free turbine turboprop engine with the characteristics shown in Table VII-1. This engine drives a 9-foot diameter, fully-reversible propeller to provide for maneuvering control when paired. These modules are completely self-contained units following the same philosophy applied in the design of the lift power plant modules. For transportation purposes these modules are divided into three parts.

- Supporting tower
- Engine nacelle
- Propeller.

These power plants also incorporate an electrical generating system for power when the craft is in the displacement mode. These modules include:

- Engine
- Propeller
- Engine cowling
- Fuel tanks and system
- Oil system
- Generator
- Engine controls

7.4.9.1 Fuel Requirements

Propulsion Power Plant

Range	50 nautical miles
Speed	40 knots
Operating time	$\frac{50}{40} = 1.25$ hour
Power Required	1,500 SHP
S. F. C.	.8

Fuel Required

Weight	=	$1,500 \times 0.8 \times 1.25$	=	1,500 pounds
Volume	=	$\dfrac{1,500}{62.36 \times 0.8}$	=	30 cubic feet
Capacity	=	30×7.48	=	224 U.S. gallons
Capacity per tank	=	112 U. S. gallons		

7.5 MODULE SUMMARY

Reference to Table VII-3 shows a total volume for the modules of approximately 14,000 cubic feet. This is a considerable achievement in reduction of the bulk. For comparison it may be noted that Case II of the sectionalized GEM (existing design) was the better of the two cases, but still reduced to only 22,700 cubic feet. The major gain accrues from the compacted lift system of the LOTS carrier designed in this study.

At this point, it is possible to consider another important facet of over-all transportability. For a vehicle dry weight of 40,000 (the final weight determination is only slightly different), the LOTS carrier represents cargo density of 2.86 pounds per cubic foot. Compared to other cargo, this is an exceptionally low density as can be seen from the cargo carrying capabilities of the following aircraft:

TABLE VII-3
SUMMARY OF MODULE DATA FOR LOTS CARRIER

Part No.	No. of Parts	Description	Length (ft.)	Width (ft.)	Height (ft.)	Volume Each cu. ft.	Volume Total cu.ft.
1	2	Cabin	9	7	7	441	882
2	2	Bow	9	7	4	252	504
3	2	Front & Rear ends	20	4	4	320	640
4	2	Ramps	20	1.5	8.5	255	510
5	2	Stern	6	7	4	168	336
6	10	Main Str.	34	5.5	4	748	7,480
7	6	L. P. P.	11	7	7	539	3,234
8	2	P. P. P.	6	3	3	54	108
9	2	Tower	12	1	1	12	24
10	2	Propeller	9	9	1	81	162
11	(4)	Wall	11	0.1	7	7.7	31
Total	32						13,981

NOTE: (Floor is not included above but will not alter practical usefulness of result).

Transport Aircraft	C-133	C-130	C-123
Cargo Compartment volume (cu. ft.)	13,900	3,700	2,600
Usable volume (0.75 packing factor)	10,400	2,780	1,950
Weight carried, normal mission, lb.	95,000	29,500	11,785
Optimum cargo density (lb/cu.ft.)	9.1	10.6	6.0

When carried by air, it is obvious that the LOTS carrier is of such a low density that transport aircraft would operate at greatly reduced capacity in terms of weight carried.

This problem of aircraft utilization can be viewed in terms of requirements for shipment of one or more LOTS carriers. Again using a packing factor of 0.75, the following aircraft requirements result for one LOTS carrier:

C-133	1.3 aircraft loads
C-130	5.0 aircraft loads
C-123	7.2 aircraft loads

The conclusion to be drawn from these data is that while each individual module of the LOTS carrier in this study is readily transportable, the aggregate bulk inhibits air transportability. For example, the C-133, capable of carrying 95,000 pounds (on a normal mission) could be loaded with only 30,000 pounds for a 31.5 per cent utilization.

7.6 A SERIES OF MODULAR VEHICLES

The present studies have not uncovered any obstacles to the building of a series of variously-sized vehicles from the modules herein designed. Several operational situations could bring about new requirements:

- Terrain requires increased operating height due to natural obstacles and excessive grades:

  solution: add two power modules with control connections;

  penalty: off-design operation of lifting system.

- LOTS carrier lost through accident or otherwise, but power modules are salvaged:

  solution: transfer power modules to remaining craft to increase their load capability;

  penalty: off-design operation and high cushion pressure.

- Loads are consistently high density:

  solution: remove one pair main structural modules;

  penalty: off-design operation and high cushion pressure.

- Loads are consistently low density:

  solution: remove one pair of power modules;

  penalty: none.

A smaller vehicle is a possibility. Four main modules and a pair of lift modules are removed. The pertinent data for the vehicle are:

Cushion Area (vs. 1,300 square feet) (square feet)	700	700	700
Lift Horsepower (vs. 5,400)	3,600	3,600	3,600
Length of Vehicle (vs. 70 feet) (feet)	48	48	48
Beam of Vehicle (unchanged) (feet)	34	34	34
Operating Height (feet)	2.5	2.0	1.5
Gross Weight (pounds)	44,000	50,000	57,000
Cargo (pounds)	5,500	11,500	18,500
Cushion Pressure (pounds per square foot)	63	71	82

The smaller vehicle is not attractive by comparison with the basic design. At equal operating heights, with 67 per cent of the original lift power it carries only 18 per cent of the original load. At reduced operating heights its cargo capability improves. However, due to its shorter length and reduced gross weight, it is, in effect, much stronger and could resist a load factor at least twice that of the original design. It is capable of greater speed as the propulsion modules, sized for a larger vehicle, remain unchanged. An advantage to transverse main modules is apparent in contemplating these modified vehicles.

The addition of two power modules and four main modules is not quite as straightforward but could also be accomplished. The vehicle would have 7,200 horsepower available to the lift system. Engine controls would need to be improvised. Placarding in order to control the load factor would be necessary but this could take the form of a control on cargo location rather than speed. In any case, it could not make the speed of the basic design.

7.7 COLLAPSIBLE COMPONENTS

The findings in regard to collapsible components are completely negative. No application of a collapsing device was uncovered. This is unfortunate since the LOTS carrier has been shown to be of such low density (almost comparable to empty boxes). However, the facts are that: the main modules must contain orthogonally disposed beams, mutually supported by structural joints and must furthermore be sealed at all exterior seams so as to provide buoyancy; several of the miscellaneous modules are in the same category structurally as the main modules; the remaining modules, e.g., power and cabins, are not of an extremely low density; finally, collapsible components must cause an additional weight penalty over and above that of modularization and would severely detract from the utility of a GEM.

7.8 INFLATABLE COMPONENTS

The application of inflatables is studied in Reference 26, a companion effort to this study. Full details are contained there which show most unattractive size requirements for a structure to perform the function of a GEM.

However, there are some devices which are essential to the success of a lightweight structure operating in a "high load" environment. These are fenders and bumpers for the sidewalls and a cushion for the cargo deck. Since these are auxiliaries which one would expect to procure from shelf sources, no detailed design is executed in this study. Minimum weight only is all that could be tolerated and inflatables should offer maximum cushioning per pound of weight.

CHAPTER VIII

STRUCTURAL DESIGN OF MODULES

8.1 SYMBOL LIST

M	Bending moment - inch-pounds
V	Shearing force - pounds
t	Thickness - inches
d	Diameter or typical spacing - inches
α	Web buckle angle - degrees
σ_u	Allowable Stress - pounds per square inch
Q	Area moment - inches^3
I	Area moment of inertia - inches^4
J_e	Section property - inches^4
u	(as subscript) - Ultimate
cr	(as subscript) - Critical
M.S.	Margin of Safety
τ	Shear stress
t	(as subscript) - Tensile or due to tension
c	(as subscript) - Crushing
p	Running load - pounds per inch

8.2 PREREQUISITES TO STRESS ANALYSIS

A general arrangement has been evolved to perform the LOTS and secondary missions. The requirements of transportability have been postulated and a modular breakdown proposed. The environment has been considered in a general way and subsequently load factors and materials evaluated. Preliminary weight and balance data were used in conjunction with the load factors to arrive at shear and bending moment values. Water pressures were estimated. At this point it is possible to perform stress analysis and in the process to incorporate specific structural members and refine the preliminary weight estimates. The actual figures for use in stress analysis are quoted here briefly:

Weights (reference Table VI-3)

Empty weight		38,032 pounds
Crew		680
Fuel		6,900
Cargo		30,000
All-up weight		75,612
Bending Moments		35×10^6 pounds per inch (reference Fig. VI-9)
Shear		192,000 pounds (reference Fig. VI-8)
Water pressure	peak	25 pounds per square inch (reference paragraph 4.8)
	general	10 pounds per square inch
Deck pressure		300 pounds per square foot (reference paragraph 4.8)

Tie-down load factors (reference paragraph 4.8)

Vertical	downward 4.0, upward 3.0
Longitudinal	forward 6.0, aft 3.0

athwartship	3.0
Maximum resultant	6.0

Materials (reference paragraph 5.6)

Sheet stock	6061-T6 Aluminum alloy, clad
Structural Shapes	6061-T6 Aluminum alloy
Forgings	6151-T6 Aluminum alloy
Rivets	6061-T6 Aluminum alloy

8.3 MAIN STRUCTURAL MODULE

The main structure has been divided into 10 transverse modules for shipping purposes having the following dimensions:

Length (L)	66 inches
Width (W)	408 inches
Height (H)	48 inches

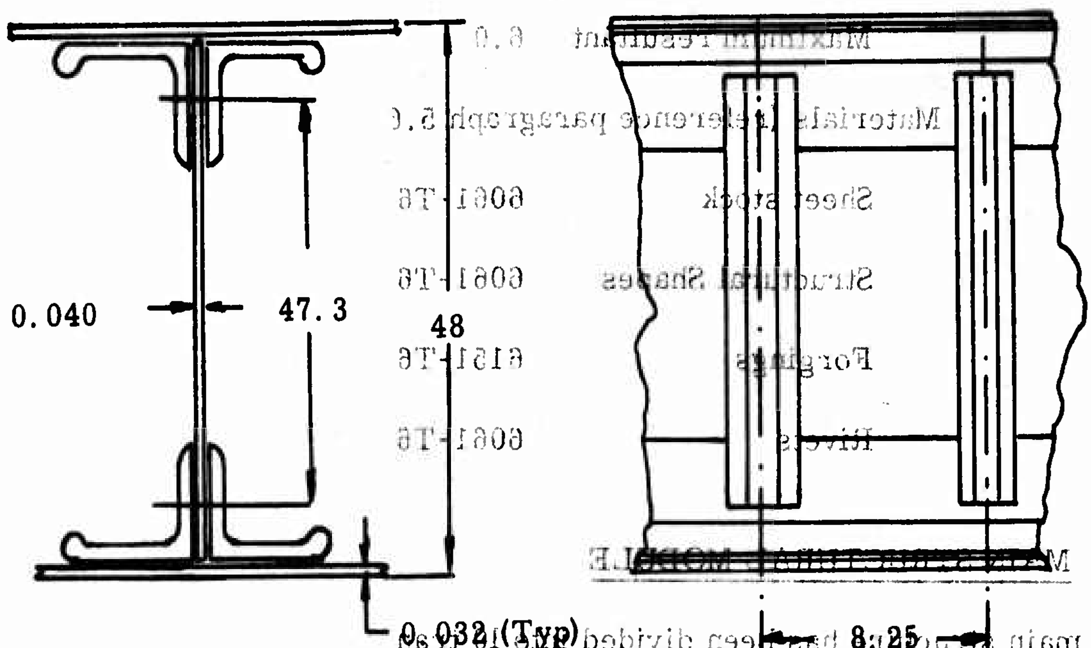
Each module is built up by 15 longitudinal beams and three transverse beams. They carry the longitudinal and transverse bending moments and shear loads. All modules have been designed to the same requirement for interchangeability regardless of whether they will be located at positions of peak shear and bending moment. The beam construction is in accordance with aircraft practice, with caps taking the tension and compression loads and the thin web taking the shear in a tension field.

Each one of the 15 beams takes 1/15 of the longitudinal bending moment and shear load.

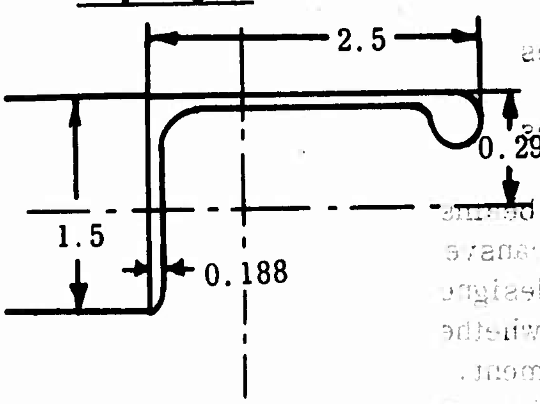
$$M = \frac{35,000,000}{15} = 2,333,333 \text{ inch-pounds}$$

$$V = \frac{192,000}{15} = 12,800 \text{ pounds.}$$

Beam Section



Cap Angles



Section Properties

$$A = 0.846 \text{ inch}^2$$

$$I_{xx} = 0.173 \text{ inch}^4 \quad (r = 0.453 \text{ inch}$$

$$I_{yy} = 0.643 \text{ inch}^4 \quad (r = 0.872 \text{ inch})$$

$$W = 0.995 \text{ pounds per foot}$$

8.3.1 Web Stresses

Shear stress prior to buckling is very low and will not be quoted. The ultimate shear load (V_u) is found below per Sechler and Dunn (Reference 27), p. 244 and the following properties of the web:

Effective height (h)	=	47.3 inches
Web thickness (t)	=	0.040 inch
Web stiffener spacing (d)	=	8.25 inches
Rivet factor (C_r)	=	0.67
h/d ratio	=	5.73
Web buckle angle (α)	=	41.5^{o}
Stress distribution factor (R)	=	0.95
Allowable stress (σ_u)	=	38,000 pounds per square inch
Flange area moment (Q)	=	50.1 inches^3
Flange moment of inertia (I)	=	1,900 inches^4

Critical Shear Stress:

$$\tau_{cr} = K \frac{\pi^2 E t^2}{10.92 d^2} = 1,135 \text{ pounds per square inch}$$

Critical load:

$$V_{cr} = 1,135 \times 47.3 \times 0.04 = 2,150 \text{ pounds}$$

Tension field shear on web:

$$V_{tu} = \left(38,000 - \frac{1,135}{0.67}\right) 0.67 \times 0.95 \times 47.3 \times 0.04$$

$$\times\ 0.662 \times 0.75 = 21,703 \text{ pounds}$$

Total shear:

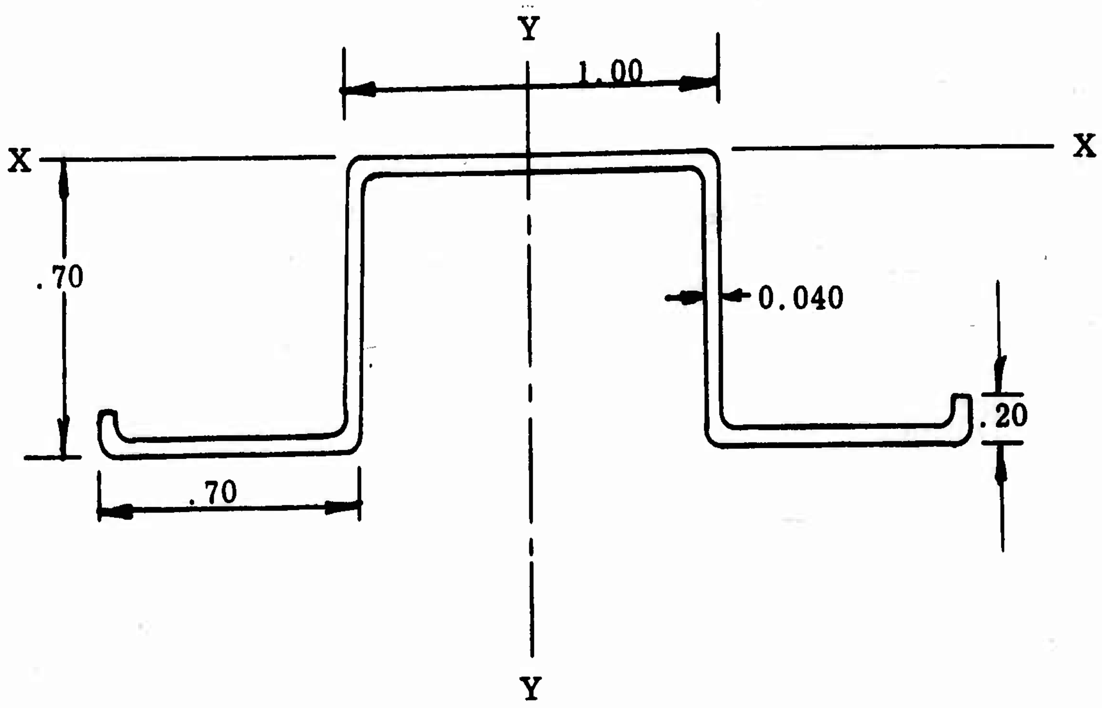
$$V_u = 21,703 + 2,150 = 23,853 \text{ pounds}$$

Margin of Safety:

$$M.S. = \frac{23,853}{12,800} - 1 = +0.86$$

8.3.2 Stiffener Stresses

Shear load is again determined (Reference 27), and stiffener values are unchanged from quoted figures in web analysis:



Determine total web shear above V_{cr} prior to stiffener failures:

$$V_{tu} = 0.023\ E\ \sqrt[3]{J_e \frac{th^2}{d}} = 22,310 \text{ pounds}$$

(J_e = 1/3 (developed width) t^3 of stiffener)

Total shear:

$$V_u = 22,310 - 2,150 = 20,260 \text{ pounds}$$

Margin of Safety:

$$\text{M. S.} = \frac{20,260}{12,800} - 1 = +0.72$$

8.3.3 Cap Stresses

The main beams are checked for direct stress in the caps (σ_c) and crushing on the compression flange (σ_{cc}), plus the bending moment due to the tension field. (Refer again to Sechler and Dunn, p. 256 .)

Direct Stress

$$P = \pm \frac{M}{h} - 0.5\, V_t \cot \alpha \quad \text{where} \quad V_t = V - V_{cr}$$

$$= \pm\ 49,330 - 5,647 = -\ 54,977 \text{ pounds} \quad \text{(compression)}$$

$$\sigma_c = \frac{54,977}{1.692} = 32,492 \text{ pounds per square inch.}$$

Margin of Safety:

$$\text{M. S.} = \frac{38,000}{32,492} - 1 = +\ 0.16$$

Crushing Strength of Caps

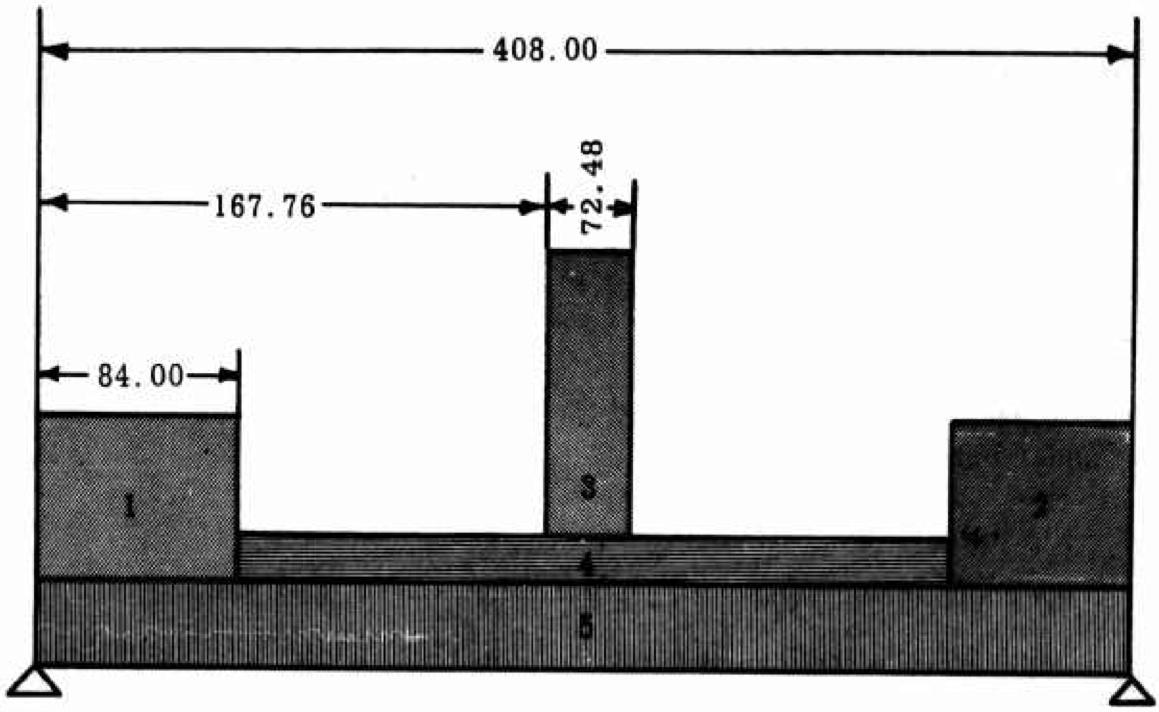
$$\sigma_{cc} = \frac{P_{cc}}{A}$$

$$\sigma_{cc} = \frac{10,650 + 17,200}{0.864} = 32,300 \text{ pounds per square inch}$$

$$\text{M. S.} = \frac{32,300}{32,492} - 1 = -.006$$

8.3.4 Transverse Beams Web Stresses

It is assumed that the critical case is the group of beams passing under the central power module. The bending moment is computed from this loading diagram:



$p_1 = p_2 = 33.14$ lb./in.

$p_3 = 275$ lb./in.

$p_4 = 1.83$ lb./in.

$p_5 = 7.92$ lb./in.

$$M_1, M_2 = 116,917 \text{ in./lb.}$$

$$M_3 = 1,852,480 \text{ in./lb.}$$

$$M_4 = 34,416 \text{ in./lb.}$$

$$M_5 = 164,799 \text{ in./lb.}$$

Total Bending Moment:

$$M = 2,168,612 \text{ in./lb. (for 1.0 load factor)}$$

$$\text{Shear } V = 14,605 \text{ lb.}$$

For each of 5 beams and a load factor of 6.0, the moment and shear become:

$$M = \frac{6 \times 2,168,612}{5} = 2,602,334 \text{ in./lb.}$$

$$V = \frac{6 \times 14,605}{5} = 17,526 \text{ lb.}$$

The computation of stress is similar to that given for the longitudinal beams. The transverse beams are loaded to a lesser degree but material gage is less and the spacing is altered. Using the same configurations as was used for longitudinal beams, i.e., the same caps and stiffeners but reducing the web to 0.032 inch:

h 47.3 inches

t 0.032 inches

d 7.5 inches

$$C_r = 0.67$$

$$h/d = 6.31$$

$$\alpha = 42.61^{o}$$

$$\sigma = 38,000 \text{ pounds per square inch}$$

$$Q = 46.07 \text{ in.}^3$$

$$I = 2,183 \text{ in.}^4$$

$$K = 5.35$$

Critical Shear Stress:

$$\tau_{cr} = \frac{5.35 \times 9.87 \times 10^7 \times .001024}{10.92 \times 56.25} = 880 \text{ lbs./in.}^2$$

Critical Load:

$$V_{cr} = 880 \times 47.3 \times 0.032 = 1,332 \text{ pounds}$$

Tension Field Shear on Web:

$$V_{tu} = \left(38,000 - \frac{880}{0.67}\right) 0.67 \times 0.94 \times 47.3 \times 0.032 \times 0.6775 \times 0.737$$

$$= 17.463 \text{ pounds}$$

Total Shear:

$$V = (17463 + 1332) = 18,795 \text{ pounds}$$

Margin of Safety:

$$M.S. = \frac{18,795}{17,526} - 1 = + .07$$

Stiffener and Beam Cap Stresses:

The details of the computation are identical in form to those for the longitudinal beams and will not be repeated. The Margins of Safety are:

$$\text{Beam caps} \begin{cases} \text{Direct Stress:} & +0.38 \\ \text{Crushing:} & +0.12 \end{cases}$$

8.4 BOTTOM PLATING

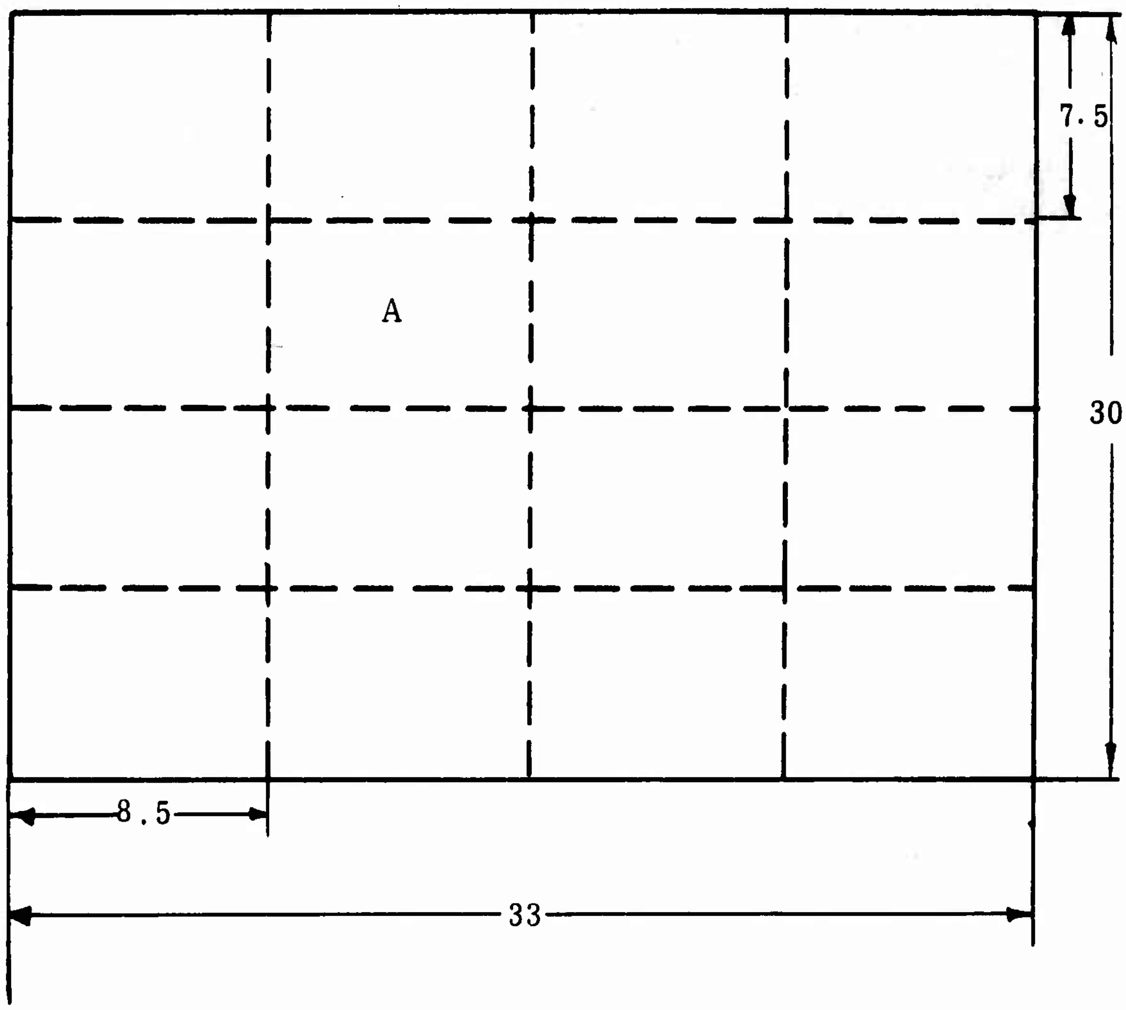
The water pressure on large bottom areas is 10 pounds per square inch. Consider a section of bottom plating as shown in the sketch. Stiffeners are indicated by the dashed lines. A single panel such as the one marked "A" is assumed to be rigidly supported and the stresses therein will be determined. The method of Sechler and Dunn (p.293) is employed. It presupposes a thin plate theory. In this range of thickness other methods are available and a wide spread in results is sometimes encountered. Another source of error in plating stress is the questionable end conditions; hence a high Margin of Safety should prevail.

Employing the symbols of the reference:

$a = 8.75$ inches $\qquad \frac{a}{b} = 1.1$

$b = 7.5$ inches $\qquad \frac{b}{t} = 235$

$t = 0.032$ inch $\qquad \frac{b}{2t} = 118$



With these values the appropriate graphs of the reference indicate the deflection (w):

$$\frac{w}{t} = 4.7 \qquad w = 0.1505 \text{ inch}$$

Also, by graphical data the stresses are obtained in both directions, the higher being in line with the short dimension of the panel:

$$\sigma = 18{,}000 \text{ pounds per square inch}$$

Margin of Safety: high

8.4.1 Bottom Stringers

Refer again to Sechler and Dunn (p. 297) and, for simplicity, use the case of a square array of stringers. The procedure commences with an allowable stress and results in a required section modulus (I/c) as follows:

$$I/c = \frac{pb^3}{16n} = 0.160 \text{ in.}^3 \text{ (for pinned end condition)}$$

$$I/c = 2/3 \times 0.160 = 0.106 \text{ in.}^3 \text{ (estimated for actual end conditions)}$$

where b is the over-all length - 33 inches and n is the number of bays - 4. For the selected stringer (bulb angle 2 x 2 x 0.125 inch) the required properties are these:

$$I = 0.217 \text{ in.}^4$$

$$I/c = 0.190 \text{ in.}^3 \text{ (to be compared to } 0.106 \text{ in}^3\text{)}.$$

8.5 MAIN STRUCTURAL MODULE JOINTS

All joints are designed to resist the maximum load condition in order to make the modules identical and interchangeable. There are, of course, locations where a lesser strength would suffice and consequently some dead weight is the penalty for the convenience in assembly. See Figure VII-4 for the fitting arrangement.

To compensate for clearance and the effect this sometimes has in permitting hard bottoming and load amplification, a fitting factor of 2.0 will be used.

Load transmitted by the flanges:

P_c = 54,967 pounds; P_t = 43,693 pounds

Forging properties (6151-T6 Aluminum Alloy):

F_{tu} = 44,000; F_{su} = 28,000; F_{br} = 92,000

(all pounds per square inch)

Bushing properties (8630 steel):

F_{tu} = 125,000; F_{su} = 75,000; F_{br} = 175,000.

(all pounds per square inch)

Bolt properties (7/8 inch diameter high tensile steel)

F_{tu} = 100,000 pounds; F_{su} = 65,000 pounds

Design Fitting Loads:

General - 54,967 x 1.15 = 63,212 pounds

Bearing - 54,967 x 1.5 = 82,451 pounds
(bushing on fitting)

Bearing - 54,967 x 2.0 = 109,934 lb.
(bolt on bushing)

Tension - 43,693 x 1.25 = 54,616 lb. (also tear out)

Shear - 54,967 x 1.25 = 68,709 lb.

Bolt in Double Shear:

$$\text{M. S.} = \frac{78,170}{68,709} - 1 = +\ 0.14$$

Bearing of Bolt on Bushing:

$$f = \frac{109,934}{0.875 \times 1} = 125,000 \text{ pounds per square inch}$$

$$\text{MS} = \frac{175,000}{125,000} - 1 = +.40$$

Bearing of Bushing on Fitting

$$f = \frac{63,212}{1 \times 1} = 63,212 \text{ pounds per square inch}$$

$$\text{MS} = \frac{92,000}{63,212} - 1 = +\ .46$$

Edge Distance

$$e = \frac{54,616}{2 \times 28,000 \times 1} = .975 \text{ (use 1.00 in.)}.$$

Tear Out

$$f = \frac{54,616}{2 \times 1 \times 1} = 27,308 \text{ pounds per square inch}$$

$$\text{MS} = \frac{28,000}{27,308} - 1 = +\ .03$$

Radius Required

$$R = 0.4375 + .0625 + 1.00 = 1.500 \text{ inches}$$

Tensile Stress

$$f = \frac{54,616}{(3 - 1) \times 1} = 27,308 \text{ pounds per square inch}$$

$$MS = \frac{44,000}{27,308} - 1 = +.61$$

8.6 BOW DESIGN

The bow has been designed according to seaplane hull design practices. Double curvature is avoided where possible for fabrication ease. The lines are shown in Figure VIII-1.

The bow consists of a keel, frames spaced every 24 inches and longitudinal stringers to take care of the bottom pressure loads plus the compression loads due to bending moments.

8.6.1 Bottom Pressure Distribution

The bottom pressure distribution has been found by the method given in MIL-A-8629 (Aer).

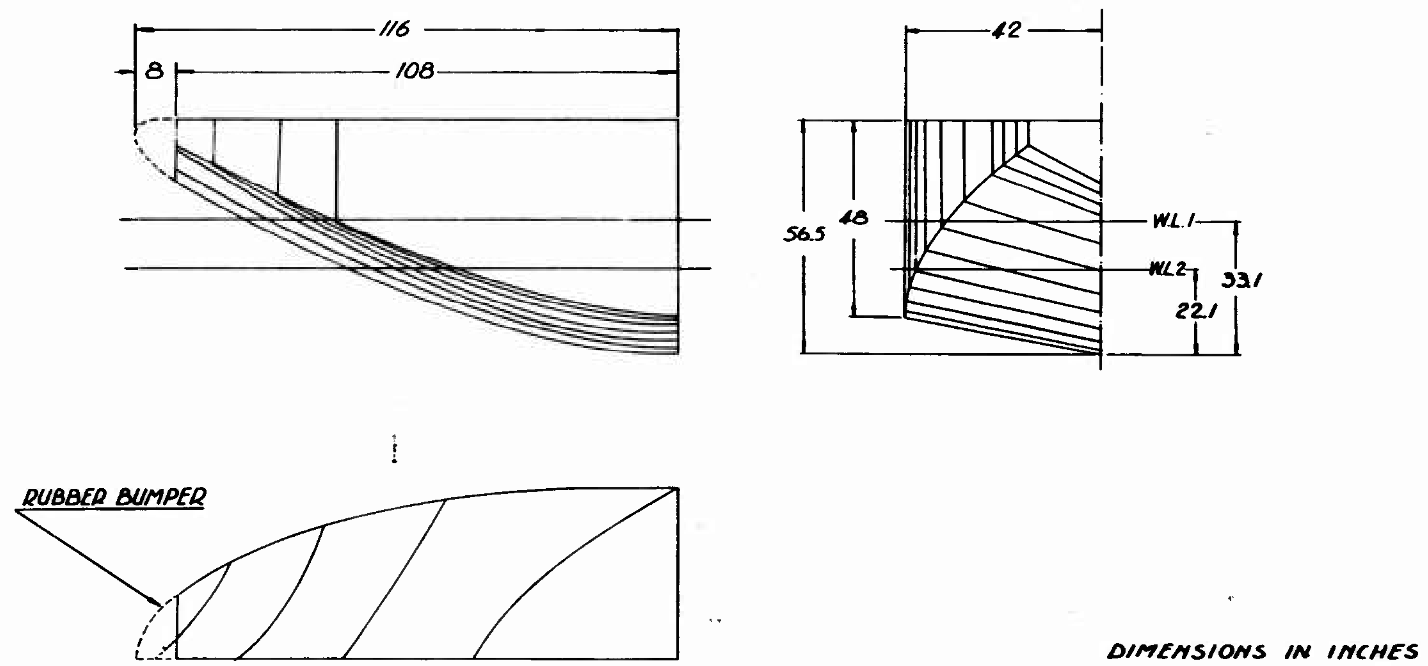
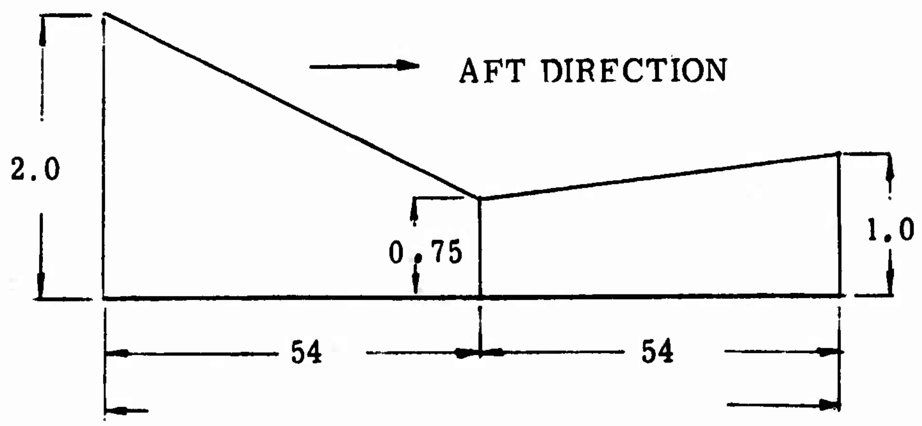


FIG VIII-1 BOW LINES FOR PRESSURE COMPUTATION



Bow Length - Analogous Distance to Step

VARIATION OF FACTOR K_1

Local pressure (p):

$$p = \frac{f \times K_1 \times V^2}{\tan \beta}$$

where:

V is horizontal speed in knots - use 40

K_1 is obtained from graph

$$f = \left.\begin{matrix} 0.00319 \text{ (keel)} \\ 0.00239 \text{ (Chine)} \end{matrix}\right\} \text{rough water operations}$$

$$\beta \text{ (deadrise angle)} = \begin{matrix} 31.15^\circ & \text{(station 0)} \\ 19.70^\circ & \text{(station 54)} \\ 11.45^\circ & \text{(station 108)} \end{matrix}$$

	Station					
	0		54		108	
	Chine	Keel	Keel	Chine	Keel	Chine
p (psi)	16.88	12.65	10.63	7.97	24.65	18.50

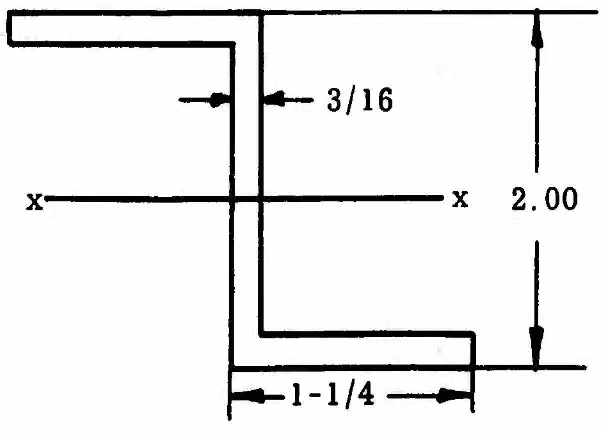
Assume that the most representative portion of the bottom is the one near the step. This will be supported by frames every 24 inches and longitudinal stringers every 10.7 inches. The design pressure is 25 pounds per inch. Bottom plate stresses for a thickness value of .080 inch (per Sechler and Dunn, p. 293) are:

$$\sigma = 32,000 \text{ pounds per square inch}$$

$$\text{M.S.} = \frac{42,000}{32,000} - 1 = +\ .31.$$

8.6.2 Stringers

Assume the stringers are continuous beams with an extruded section as sketched below and simply supported at the ends. The stresses are determined in the same manner as used for previous stringer calculations.



$$I_{xx} = 0.458 \text{ in}^4$$

$$S = 0.458 \text{ in}^2$$

$$\text{Weight} = 0.92 \text{ pounds per foot}$$

$$\sigma = \frac{16,550}{0.458} = 36,000 \text{ pounds per square inch}$$

$$\text{M.S.} = \frac{38,000}{36,000} - 1 = +.06.$$

8.7 LIFT POWER PLANT MODULES

The lift power plant modules are of prismatic shape, with the following dimensions:

L = 132 inches

W = 84 inches

H = 84 inches

Each module is a self-contained unit and comprises the power plant engine accessories, fan, ducting, fuel tanks, and controls.

The individual weights of the various items and the c.g. position are listed in Table VIII-1. They are attached to the main structural modules by fittings and bolts in the four lower corners.

The structural members are stressed for longitudinal and transverse loading cases. The load factor used is 6.0 (resultant).

The cross-section and structural layout are depicted in Figures VI-9 and VI-10. The fuel tanks are integral with the front and rear cross beams.

TABLE VIII-1
LIFT POWER PLANT MODULE WEIGHT AND C.G. LOCATION

	W Pounds	"z" Inch	W. z Inch—Pounds
Structure	7,399	363	26,858
Engine	495	32	15,840
Fan	170	71	12,070
Fuel System	116	71	8,236
Accesories	1,167	18	2,100
Mount	50	27	1,350
Exhaust	100	37	3,700
Miscellaneous	60	30	1,800
	18,476		71,954

Note: "z" distance is measured from base at module.

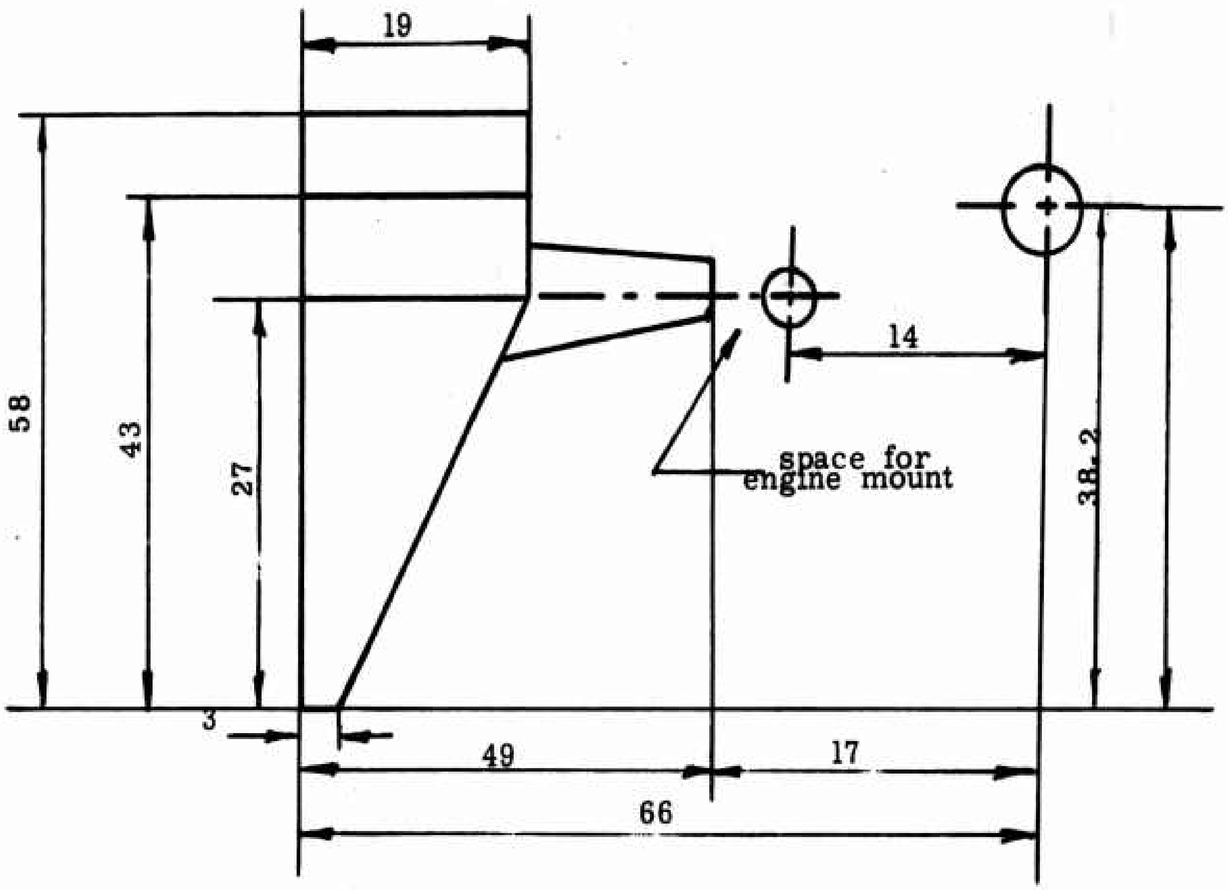
8.7.1 Engine Mounts

The load on the mounts is derived from the engine weight of 932 pounds and the tie-down load factor. Use of factors of 5.0 in the horizontal direction and 3.3 in the vertical direction (resultant is 6.0) produces the critical condition.

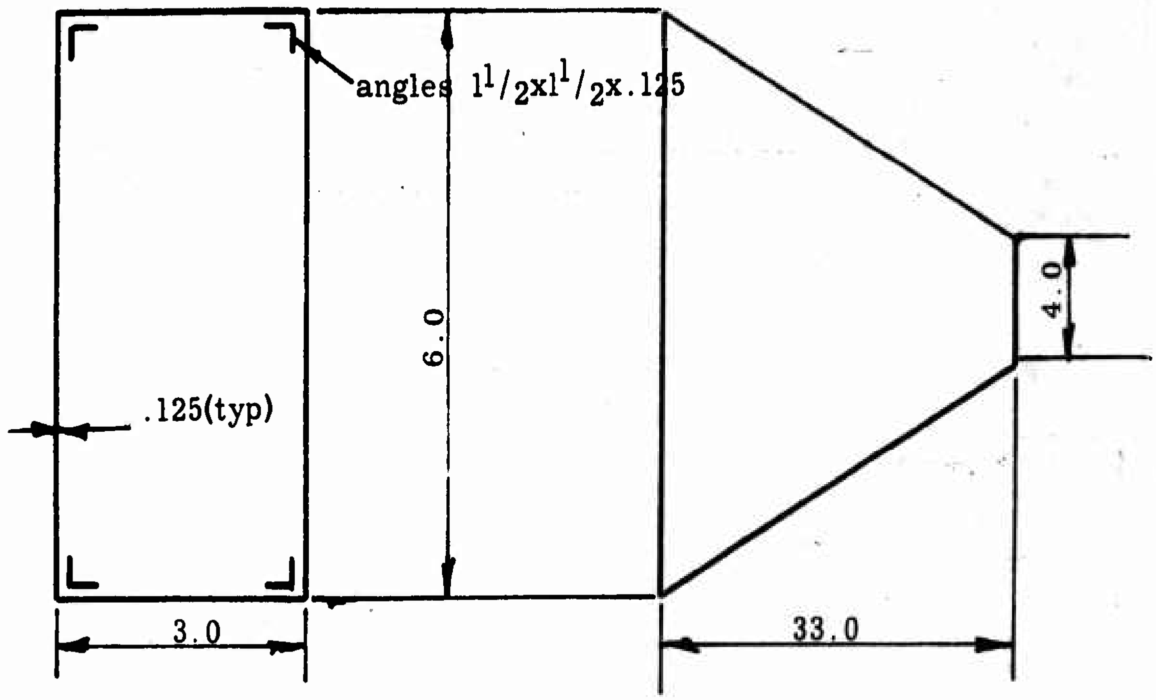
8.7.1.1 Loads

P = 3,104 pounds (vertical); P = 4660 pounds horizontal). The mounts take the form of beams as sketched below. Vertical loads are resisted by beam bending and horizontal loads (fore and aft direction) are taken by the beams so aligned, one acting as a beam column and

the other by tension. There is a small eccentricity which superposes bending on the beams in direct stress as the engine mounts are displaced from the plane containing center of gravity.



Beam sections can be seen in the sketch below which also contains the section properties:



8.7.1.2 Section Characteristics

Angle characteristics (1 x 1 x .125) (weight 0.28 pounds per foot).

$$A = 0.23 \text{ inch}^2$$

$$I = 0.0208 \text{ inch}^4$$

$$k = 0.298 \text{ inch}$$

$$I/c = 0.0293 \text{ inch}^3$$

For the composite section the pertinent properties are

$$I = 14.22 \text{ inches}^4; \quad A = 2.54 \text{ inches}^2.$$

8.7.1.3 Stress Determination

$$f = -\frac{P}{A} - \frac{Mc}{I} = -\frac{2330}{2.54} - \frac{85,546 \times 3}{14.22} = 18,965 \text{ pounds per square inch}$$

$$M.S. = \frac{1}{.026 + 0.515} - 1 = +0.83.$$

The short column condition requires checking and the allowable stress is:

$$f = 35,000 \times \frac{35,000^2 \times 33.56^2}{4 \times 1 \times \quad \times 10^7} = 31,500 \text{ pounds per square inch}$$

$$M.S. = \frac{31,500}{18,965} - 1 = +0.66.$$

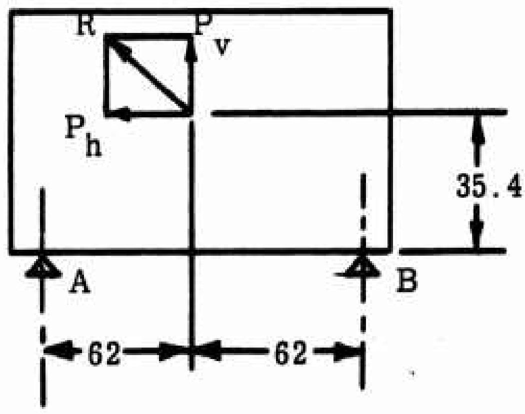
There are four supports radially disposed as previously sketched in Figure VII-10, but since the transverse beams are shorter, experience lesser loads, and have the same section properties, they do not require checking.

8.7.1.4 Power Plant Box Beams

The main structural members of the power plant module are box beams disposed peripherally about the module (see Figure VII-9). They are subjected to bending moments and direct loads. Analysis shows the Margins of Safety to be high and the details will not be reported. For example, due to a shear on the web of the front beam of 79.4 pounds per inch, a stress of 3,760 pounds per square inch resulted.

8.7.1.5 Attachment of Power Plant Module to Main Structure

The power plant module is attached to the main structure by means of four bolted joints:



Weight of module (loaded with fuel):

$$W = 932 + 900 + 625 = 2,457 \text{ pounds}$$

Resolving the loads in horizontal and vertical components (resultant is 6.0).

$$P_H = 5.37 \times 2457 = 13,194 \text{ pounds,}$$

$$P_V = 2.69 \times 2457 = 6,609 \text{ pounds.}$$

Vertical reactions

$$\Sigma M = 0 = 6,609 \times 62 - 13,194 \times 35.4 + R_A \times 124$$

$$R_A = 462 \text{ pounds (for two athwartship joints)}$$

$$\Sigma F_V = 0 = 462 + 6609 + R_B$$

$$R_B = 7,071 \text{ pounds for two athwartship joints.}$$

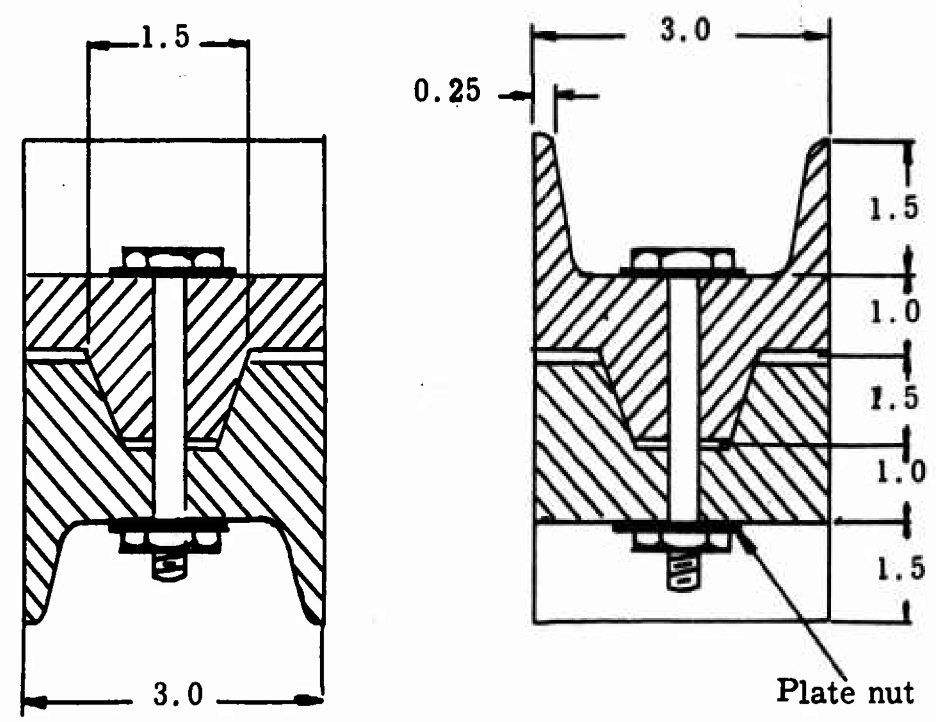
Horizontal reactions

$$R_A = R_B = \frac{13,194}{2} = 6,597 \text{ pounds.}$$

Due to the eccentricity of the load, the reaction at joint "B" is the critical case and the only one requiring analysis. The loads per joint are half of the reactions just found:

$$P_V = 3,536 \text{ pounds (upward)}; \quad P_H = 3,299 \text{ pounds (forward)}.$$

The vertical load will be supported by a bolt in tension and the horizontal load by the fitting in shear.



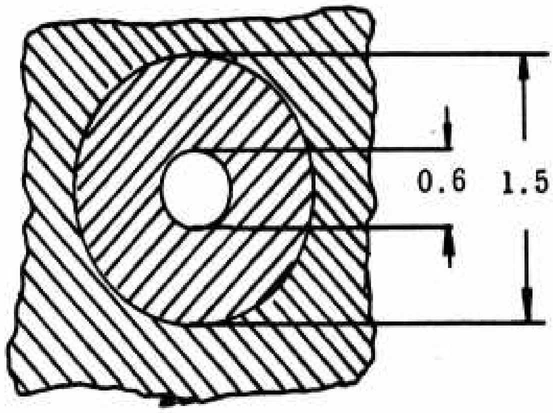
Bolt Tension

Tensile Strength of 1/2 inch bolt (high tensile): 7,400 pounds

$$M.S. = \frac{7400}{3536} - 1 = +\ 1.09$$

Fitting in Shear

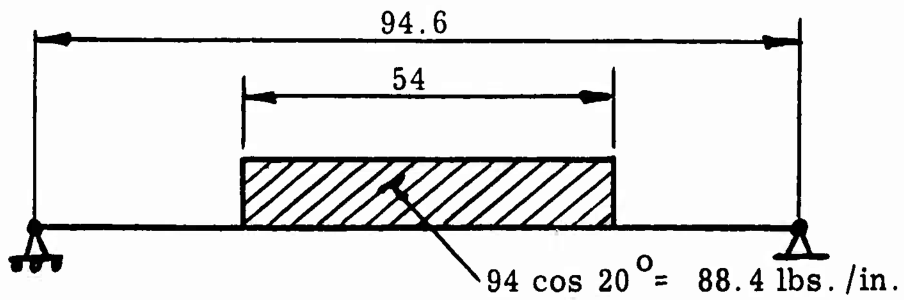
Material 6151 - T6 Aluminum Alloy



Fitting factor: 1.2, Shear stress: 4,800 pounds per square inch, M.S.: Very high.

8.7.1.6 Ramp

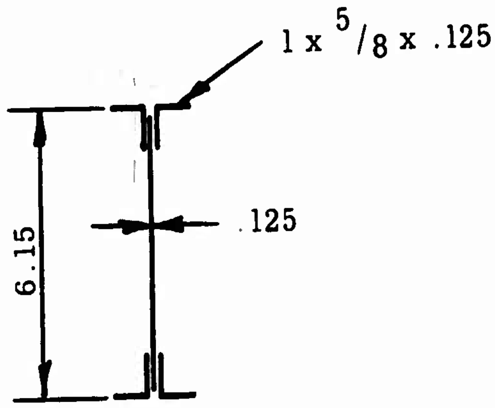
To obtain the critical condition, estimate that 75 per cent of the total load acts on the rear axles and that a load factor of 1.5 is adequate. The design load is therefore 25,400 pounds. A reasonable distribution for the load considering axle spacing and tire size, is given in the sketch:



This produces a bending moment;

$$M = \frac{88.4 \times 54}{2}\left(\frac{94.6}{2} - \frac{54}{2}\right) = 80{,}000 \text{ inch-pound.}$$

The ramp inclines to the ground at 20°. It consists of 9 longitudinal beams and 3 transverse beams, and a decking of 20 I-beams with sheet covering. The beam section properties are given in the sketch:



$I = 8.06 \text{ in.}^4$

$A = 1.53 \text{ sq. in.}$

$$I = \frac{.125 \times 6.15^3}{12} + 4 \times (.19) \times 2.724^2 = 8.06 \text{ in.}^4$$

$$A = 4.(.19) + (.125)(6.15) = 1.53 \text{ in.}^2.$$

The stress, considering that the beams resist the entire load and that a direct stress component exists due to the inclination, is:

$$\sigma = \frac{80,800 \times 3.075}{8.06} + \frac{32.2 \times 54}{1.53} = 33,470 \text{ pounds per square inch.}$$

$$= \frac{38,000}{33,470} - 1 = +0.13$$

The deck beams are athwart the longitudinals and must span only 30 inches. The I-section (a 2-1/2 x 2 x .125 aluminum alloy) has a moment of inertial of 0.831 in.^{4} and the stress is:

$$\frac{MC}{I} = \frac{17,500 \times 1.25}{0.831} = 26,400 \text{ pounds per square inch.}$$

$$M.S. = \frac{38,000}{26,400} - 1 = +0.44$$

8.8 SUMMARY REMARKS ON STRESS ANALYSIS

While the stress analysis reported in this chapter has not been carried out to the utmost detail, it does present an over-all picture of a structurally adequate design. One very slightly negative Margin of Safety was uncovered which could be corrected without perceptively altering the weight figures.

Weights derived from the sections and thicknesses reflect the design criteria as employed in this study. It may appear that some of the margins are excessively high and that a reduction in weight is possible. However, vibration effects would need detailed study which could not be included in the scope of this program and very often such detailed studies result in many structural members being "stiffness designed," regardless of whether or not the stresses are safe. In short, it is estimated that if any weight were to be pared as unjustified on a stress basis, it could go back in for other reasons and the weights as determined in this report validated.

CHAPTER IX

THE PROBLEMS OF FIELD APPLICATION OF THE MODULAR DESIGN

9.1 TRANSPORTATION TECHNIQUES

A detailed study of the problems, methods, and procedures of performing shipments is not within the scope of this report but the subject must be surveyed to determine any difficulties inherently present due to a particular LOTS carrier design.

9.1.1 Handling Provisions

The modular concept permits complete or partial knockdown. Where mechanized handling equipment is unavailable, as for example when a vehicle becomes disabled in the field and must be evacuated to maintenance shops, manual handling is entirely feasible. The main structural modules have a striking resemblance to the pontons of floating bridges and could be handled similarly. Carrying handles may be necessary and they could be disposed either on the bottom so as to do additional duty as skids or they could be placed in recessed wells so as not to interfere when the long faces of the modules about each other These main structural modules, at just under one ton (final weight determination = 1,952 pounds) are the major items. The lift power plant modules at 664 pounds have adequate locations for placement of handles.

9.1.2 Crating Requirements

For practically all the transportation modes to be encountered, no crating requirements are anticipated. The one possible exception is in the case where the LOTS carrier might become hold cargo of an ocean-going ship and would need to be protected from rough handling of this environment.

9.1.3 Tie Down Provisions

The simple lifting handles which are provided in sufficient numbers so as to give a one-man load per handle are ideally suited to use in tie down aboard any carrier. No additional provisions are required.

9.2 ECONOMY IN SHIPMENT

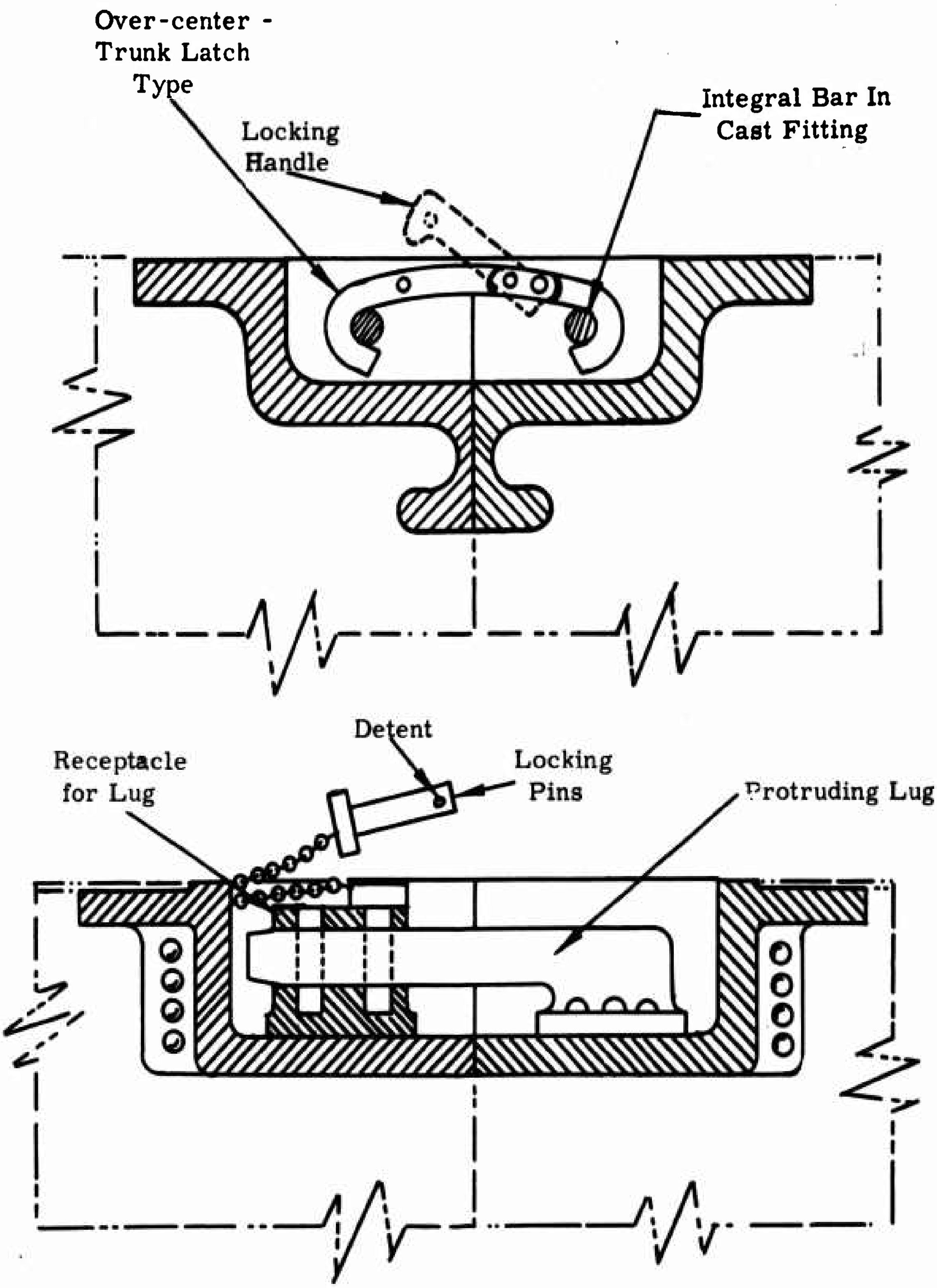
Details of the economics have not been considered but are available by extrapolation from the data contained in Reference 28 which is a comprehensive treatment of Army aircraft shipment problems. The problem of transport aircraft utilization has been discussed in connection with the density of the LOTS carrier as a cargo. Furthermore, in selecting module proportions, the problem of getting a reasonable match to cargo compartments was considered and none of the modules force a wastage of space.

9.3 FIELD ASSEMBLY

9.3.1 Main Structural Joints

For purposes of over-all design studies and weight determinations bolted joints with forged fittings have been used. In order to realize an assembly time compatible with air landing operations,quick connecting devices would be needed to be employed. Consider that 230 connections are required for the main structural modules (15 longitudinal beams connected top and bottom in 9 joints). A grand total of 400 connections is estimated. Several possible approaches to the quick acting connecter are sketched in Figures IX-1 and IX-2. In one case there are round bars cast integral in the end fittings and an over-center, trunk type latch quickly engages the bars and then goes into the latched position. Tensile preload can be quite high, generating frictional forces between modules for shear transmission. Alternatively, an alignment and shear device as sketched may be necessary. A device such as the protruding lug-retaining pin arrangement as sketched provides for direct loads and shear transmission concurrently but does not lend itself to preload across the joint.

Example "A"



Example "B"

Fig. IX-1. Latching Devices

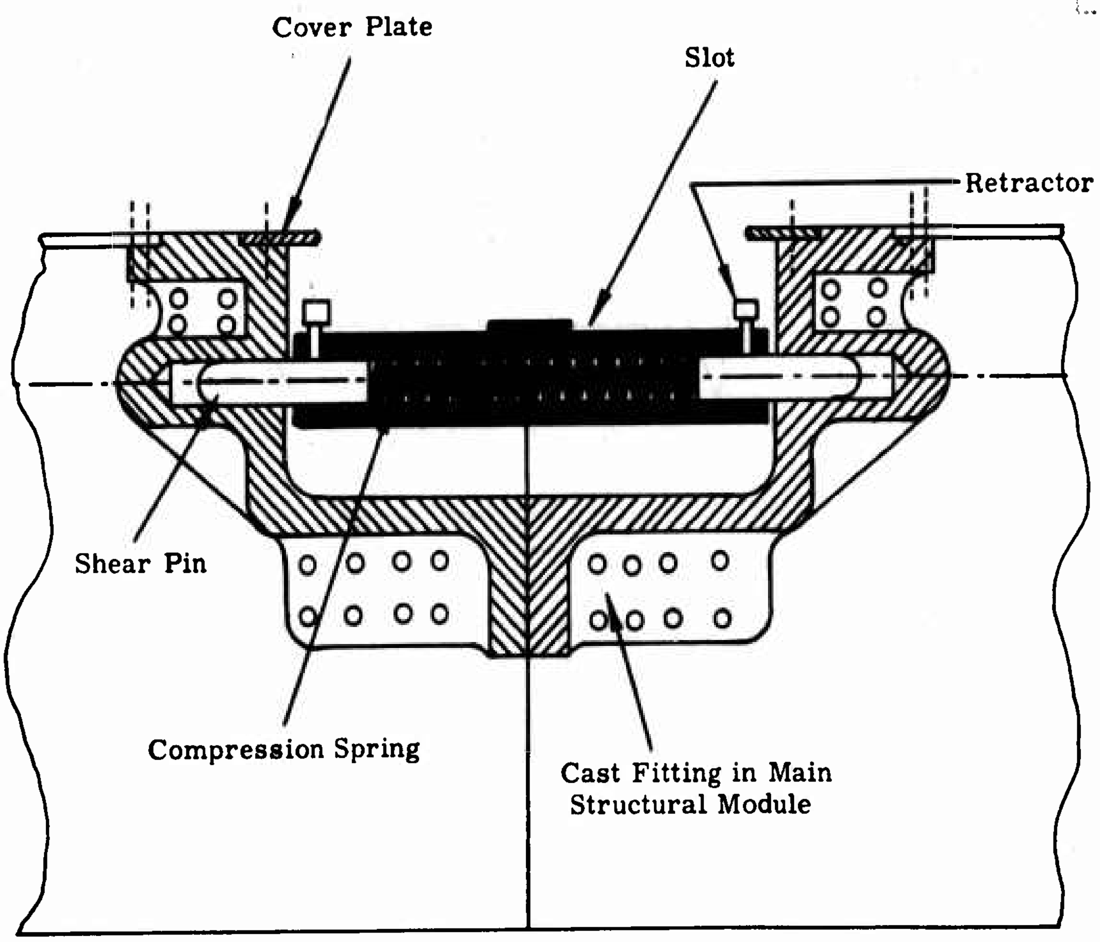


Fig. IX-2. Alignment and Shear Device

9.3.2 Other Assembly Details

There are a number of assembly problems which would require detailed engineering attention. These have been examined to determine that no insurmountable obstacle is present. The following have been considered:

- Connectors in hydraulic and mechanical control links.
- Electrical connectors to link batteries and instrument lines.
- Sealing of the peripheral annular jet.

In the case of the first two, the problem is not unique and quick-acting connecters of a suitable type are in wide use and available as stock items.

9.3.3 Assembly Procedure and Time Requirement

The assembly task would need to be a properly sequenced, standardized operation in order to consume minimum time. It would have many operations in common with other military field problems such as bridge assembly and vehicle first echelon maintenance. A procedure is suggested in Table IX-1 and times are estimated. The table is based on incorporation of the quick-acting connecters but otherwise the times are considered conservative. A four-man team (crew size) and powered handling equipment are presupposed.

TABLE IX-1
ASSEMBLY PROCEDURE

Operation	Equipment	Weight (pounds)	Man Power	Time Minutes
1. Arrange modules and parts on ground for expedited assembly	Mobile crane or fork lift truck		4	40
2. Join Main Modules 1 and 2 (a) move in position (b) level and align (c) install latching and alignment devices	Crane or fork-lift, jacks, shims, wrenches	1900 ea.		15
3. Join succesively main modules 3-10 similarly to operation 2.	Same as in "2"	Same as 2	4 4	80
4. Install duct joints seals, joint covers, etc.	Hand tools		4	15
5. Join lift power modules 1-6; steps similar to those in "2" plus hook-ups	Same as in "2"	1848 ea.	4	90
6. (See #4)			4	15
7. Join front end module	Same as in "2"	625	4	10
8. Join bow modules 1 and 2	Same as in "2'	300 ea.		15
9. Join cabin modules 1 and 2	Same as in "2"	1010		20
10. Join rear end module	Same as in "2"	625		10
11. Join stern modules 1 and 2	Same as in "2"	305 ea.		15
12. Install towers 1 and 2		120 ea.		20
13. Install propulsion power modules 1 and 2		905 ea.		30

TABLE IX-1 (continued)

Operation	Equipment	Weight (pounds)	Man Power	Time Minutes
14. Install propulsion fuel tanks 1 and 2		130 ea.		10
15. Install ramps, fore and aft		600 ea.		20
16. Miscellaneous final details (a) Unseal batteries, fuel tanks (b) Hook-up fuel lines (c) Hook-up electrical connections (d) Hook-up control cables (e) Perform checkouts (f) Install all access covers				60
17. Perform preventive maintenance				30
				495 min. (8 hours 15 min.)

CHAPTER X

SUMMARY WEIGHT DATA

10.1 FINAL WEIGHT DATA

Final gross weight determinations are very close (3-1/2 per cent) to preliminary weights (see Table X-1) and the design is validated with regard to installed power, cushion pressure, and operating height. The most substantial difference is in the main structural module which, as designed, is 1,952 pounds or 336 pounds (21 per cent) over the initial estimate. Two factors caused this underestimate: rules-of-thumb are based on lower loads than used herein, and the estimated penalty for joints was unconservative.

The table also shows the part of the gross weight taken up by each category. The structure consumes 40.3 per cent; an evaluation of this figure appears in several locations in the report depending on the context in which it is discussed. It is not unsatisfactory for an adequately rugged vehicle. Disposable load consumes 47.3 per cent, a respectable figure. Of the disposable load 23 per cent is fuel but it may be recalled that installed power and fuel consumption are determined for limit conditions. Hence, for "normal" conditions where 2-1/2 foot ground clearance is not necessary, if fuel consumption can be halved, 3,800 pounds of additional cargo (13 per cent of capacity) can be carried.

Final weight and balance is given in Table X-2. The c.g. has been matched to the cushion center of pressure for both the loaded and unloaded cases.

10.2 DETAILED STRUCTURAL WEIGHT BREAKDOWN

The weight data for all structural elements are included in Table X-3. This is completely self-contained and warrants little discussion except to note that the over-all result closely resembles those generally encountered in aircraft practice.

TABLE X-1
COMPARATIVE STRUCTURAL WEIGHTS

Item	Original Weight	Final Weight
Main Structural Module	16, 160	19, 520
Deck	2, 390	2, 390
Bow	800	852
Stern	610	638
Front End	625	1, 648
Rear End	625	648
Lift Power Plant Module	4, 440	3, 984
Propulsion Module	260	260
Ramp	1, 640	1, 238
Tower	240	240
Cabin	614	618
Walls	164	164
Propulsion Fuel Tanks	260	260
	28, 828	31, 460

WEIGHTS BY CATEGORY

	Weight (pounds)	% of Gross Weight
Structure	31, 460	40. 31
Power Plant	7, 790	9. 99
Furnishing	1, 214	1. 56
Crew	680	0. 87
Fuel	6, 900	8. 84
Cargo	30, 000	38. 43
Gross Weight	78, 044	100. 00
Ratio $\frac{\text{Disposable load}}{\text{Gross Weight}}$		47. 28
Weight, empty	40, 464	

TABLE X-2
FINAL WEIGHT AND BALANCE

	W lbs.	x in.	z in.	Wx in-lbs.	Wz in-lbs.
Cabin(s)	2,024	68	77	137,632	155,848
Bow(s)	852	66	30	56,232	25,560
Front End	648	92	32	59,616	20,736
Front Ramp	619	59	86	36,521	53,234
Main Structure Module(s)	19,520	438	24	8,549,760	468,480
Cargo Floor Module(s)	2,390	438	49	1,046,820	117,110
Side Panel(s)	164	438	90	71,832	14,760
Lift Power Plant Module(s)	10,032	438	87	4,394,016	872,784
Rear End	648	784	32	508,032	20,736
Stem(s)	638	791	32	504,658	20,416
Propulsion Power Plant Module(s)	1,810	798	190	1,444,380	343,900
Tower(s)	240	803	162	192,720	38,880
Propulsion Power Plant Fuel Tank(s)	260	810	68	210,600	17,680
Rear Ramp	619	788	86	487,772	53,234
Weight, Empty	40,464	437.4*	549.5	17,698,591	2,223,358
Crew	680	72	74	348,960	50,320
Fuel Lift Power Plant	5,400	438	81	2,365,200	437,400
Propulsion Power Plant	31,500	810	68	1,215,000	102,000
Empty Weight	48,044	433.92	58.55	21,327,151	2,813,078
Cargo	30,000	428.5	73	12,855,521	2,190,000
Gross Weight	78,044	438*	64.1	34,183,272	

* Note correspondence of c.g. location, loaded vs. empty.

TABLE X-3
FINAL STRUCTURAL WEIGHT

Item	Material	Weight	Weight Ea. Module	No. of Modules	Total Weight
Main Structural Module	Top and bottom plating	169			
	Long, beam web	186			
	Transverse beam webs	184			
	Web stiffeners	153			
	Bottom stringers	160			
	Caps	718			
	Joints	382			
			1,952	10	19,520
Cargo Floor	Plating	101			
	Stringers	98			
	Stiffeners	40			
			239	10	2,390
Front & Rear Ends	Plating	289			
	Stiffeners	130			
	Caps	149			
	Joints	80			
			648	2	1,296
Bow	Bottom plating	60			
	Other plating	115			
	Bottom stringers	140			
	Side stringer	11			
	Caps	47			
	Joints	53			
			426	2	852
Stern	Plating	145			
	Stiffeners	54			
	Caps	78			
	Joints	42			
			319	2	638

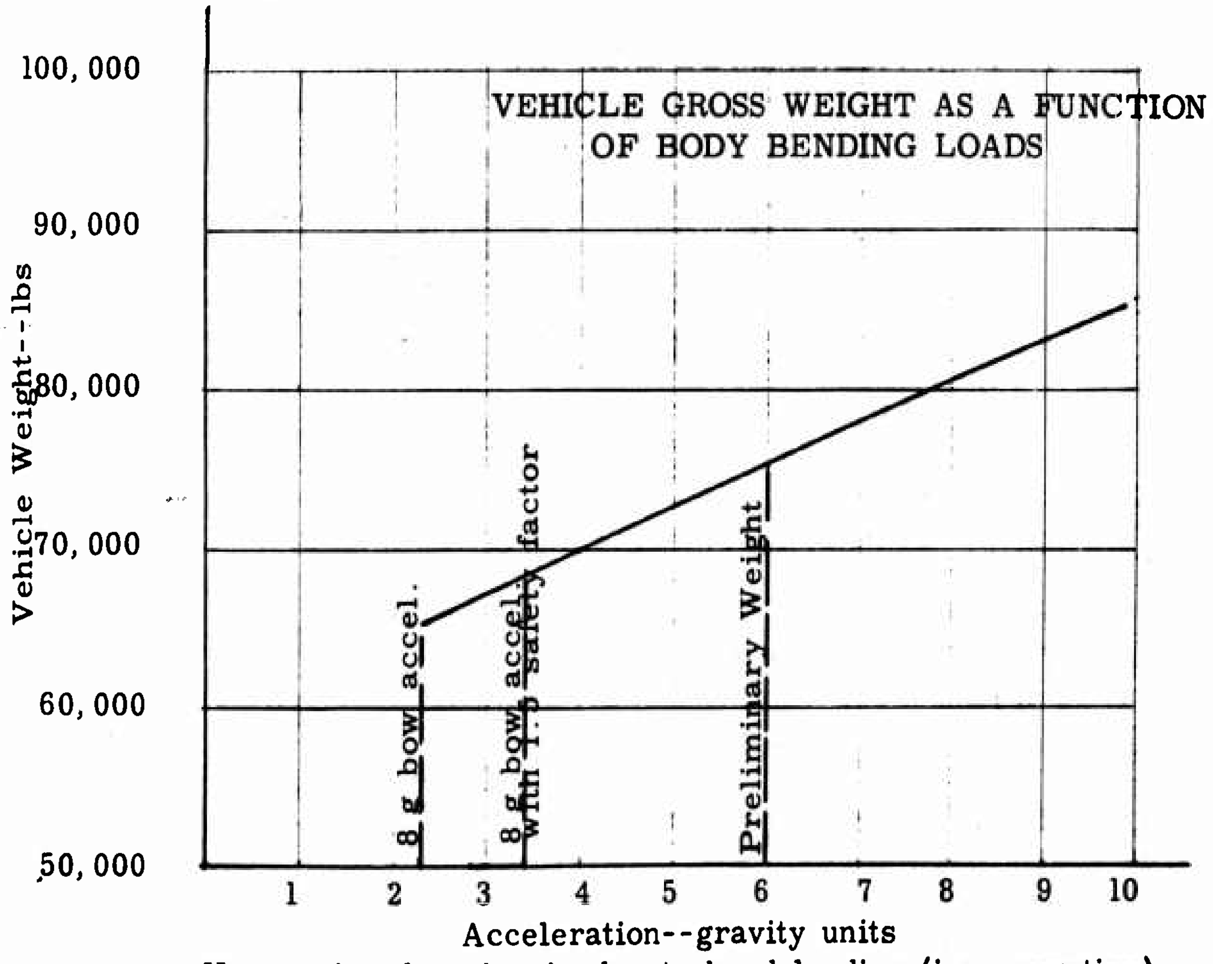
Note: Weight in pounds.

TABLE X-3 (continued)

Item	Material	Weight	Weight Ea. Module	No. of Modules	Total Weight
Cabins	Plating	76			
	Stringers	17			
	Stiffeners	21			
	Floor beams	28			
	Floor	30			
	Glass	109			
	Joints	28			
			309	2	618
Lift Power Plant Modules	Plating (general & ducting)	349			
	Plating supports	20			
	Stiffeners	76			
	Caps	169			
	Joints	50			
			664	6	3,984
Ramps	Plating (floor)	149			
	Plating (web)	71			
	Caps	68			
	Transversal beams	331			
			619	2	1,238
Propulsion Power Plant Support	Columns and tension cables	120			
	Mount and cowling	130			
			250	2	500
Side Panels	Sandwich material	41			
			41	4	164
Propulsion Power Plant Fuel Tank	Plating	130			
			130	2	260
					31,460

10.3 STRUCTURAL WEIGHT VS. STRUCTURAL CRITERIA

Since there will be an arbitrary element in the selection of load criteria until such time as considerable GEM operating experience is available, it will be helpful to have some indication of the consequences of any selection. Structural members of the main body sections are stressed in bending and as a first approximation the stress and weight are made proportional to the bending moment. See Figures VI-7, VI-8, VI-9 for the bending moment curves.



Notes--Acceleration is due to hard landing (i.e. negative), no safety factor, reactions at Stas. No. 140, 735.

10.4 STRUCTURAL WEIGHT VS. DESIGN FEATURES

Numerous mention has been made at the utilitarian aspect of the design but actually some conveniences have been retained. It will be of interest to observe the following:

Feature	Weight Saving
1. eliminate aft ramp	600 lbs.
2. eliminate both ramps	1,200 lbs.
3. eliminate aft ramp,halve front ramp	900 lbs.
4. combine cabins (following 2 or 3)	250 lbs.
5. reduce crew from 4 to 3	250 lbs.
6. reduce cabin space (following 4 and 5)	100 lbs.
7. reduce bow protuberance (with 4 and 6)	200 lbs.
8. reduce functional cargo area to 400 sq.ft.	600 lbs.

10.5 STRUCTURAL WEIGHT VS. MODULE SIZE

This data is available in Figure X-1 as a plot of weight penalty versus number of joints for the main structural modules. At 10 modules (9 joints) we suffer approximately 4,000 lbs. of weight penalty. At 8 modules, the penalty is off about 800 lbs. (at 3,200 lb.) but the formerly 5.5 foot dimension goes to 6.9 feet and fitting problems in cargo compartments are severe. In the largest transport aircraft (C-133), the cross-section of the compartment would then take only one structural module.

10.6 COST DATA

The logic of including cost data in the chapter derives from the common use of weights as an estimating device. In Reference 29, a group of approximately 20 airplanes are studied in detail and some apparently valid and definitely useful data is presented. Of course

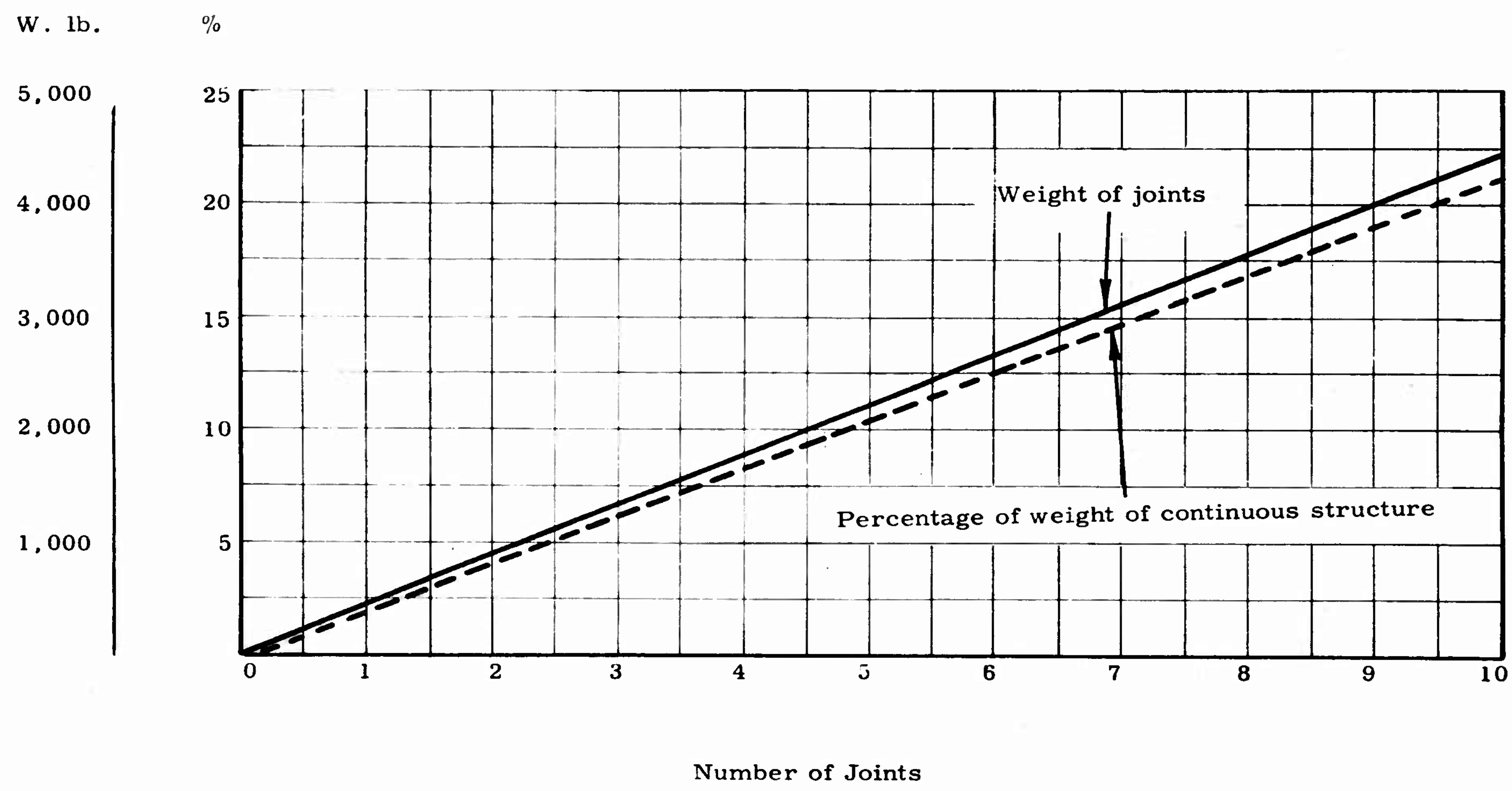


Fig. X-1. Modularization Weight Penalty -- Main Structure

the final result in dollars per pound shows a great spread. For example, for subsonic airplanes the bare airframe costs range from $40.00 per pound to $64.00 per pound for the 100,000-pound bracket, a 50 per cent variation. On this basis the bare structure for a GEM might be estimated at $30.00 per pound. The LOTS carrier bare airframe could cost one million dollars.

The effect of modularization will be small. Several tendencies in opposite directions will cancel each other. For example, where parts are interchangeable, greater precision is required than when parts are permanently joined and where small tailoring operations are allowed at assembly. This characteristic will send costs up. On the other hand, where a larger number of smaller units must be fabricated, jig and fixture costs are down and fixtures can be provided to maintain the required accuracy. Something like the degree of sophistication in quick-acting connections can have considerable effect on the final structure.

CHAPTER XI

CONCLUSIONS AND RECOMMENDATIONS

This investigation in structural design practice of Ground Effect Machines has been primarily the execution of a vehicle design for application in the Army's amphibious regime of transportation (15-ton LOTS carrier). Particular emphasis has been placed on the feasibility of attaining maximum transportability of the vehicle. As an adjunct, studies were performed on the mission of such a vehicle, the state-of-the-art in GEM structural design, the environment, structural criteria, and structural materials. A number of conclusions follow from the total investigation.

11.1 CONCLUSIONS WITH REGARD TO TRANSPORTABILITY

- It is technically feasible to produce a 15-ton LOTS carrier which is highly transportable and could be carried aboard current military air transports as well as road, rail, and ocean-going transport.

- This degree of transportability for a vehicle in the 40-ton gross weight category can only be achieved when it is an initial requirement, since designs which evolve without this requirement tend to greater bulk and present a prohibitive number of joint problems.

- A modular concept of design in which the use of identical modules is maximized and the modules are held to convenient size (just below one ton) is required for achieving this high transportability.

- The weight penalty associated with a modular design is approximately 4,000 pounds (18 per cent of the structural weight) and is the same order of weight variation as would occur within a practicable range of other design requirements.

- Numerical estimates of the time and effort to assemble a knocked-down modular LOTS carrier are not far enough off the six-hour requirement of Phase III air landings to disqualify the carrier.

- The most severe obstacle to airlifting the LOTS carrier is its low density, even when knocked down (2.86 pounds per cubic foot), which results in approximately 30 per cent utilization of transport aircraft on a weight basis at normal ranges.

- The advantages of simplified maintenance, repair by module replacement, and general transportability warrant design and construction of GEM vehicles in the 15-ton LOTS carrier category and downward on the modular principle.

11.2 CONCLUSIONS WITH REGARD TO COMPOSITE DESIGN

- As compared to the prevailing trends in GEM design, a simplified design can be evolved and applied to a 15-ton LOTS carrier -- the major simplifications being in ducting of the lift system, power distribution to the lift system from turbine engines, and generation of control forces.

- The 15-ton LOTS carrier is capable of performance generally in accord with the missions it could be expected to perform; i.e., it could operate at 40 miles per hour sustained speed and with full load it has sufficient ground clearance and range for the Army's amphibious regime of transportation.

- A variety of operating conditions can be tolerated, some of which improve the performance -- such as reduced fuel consumption at lower ground clearance -- and some of which detract from performance -- such as reduced cargo capacity at ranges greater than normal in LOTS operations.

- Particular capabilities could be emphasized by variation of the design parameters but in every case some other interrelated capability will be found to suffer -- the most glaring example of this being the case where cushion pressure would be lowered to reduce installed power and fuel consumption at the penalty of a bulkier and less transportable vehicle.

- The attempt to simplify ducting and power plant modules by a coaxial P. P. M. -- annular jet design results in a design which has a ratio of cushion area to planform area of approximately 0.6.

- The current state-of-the-art shows that bargelike shapes with minimum bow such as evolved in this design are conducive to payload-gross weight improvement as compared to vehicles having large bow protuberance.

- Attempts to further improve the ratio were not pursued to fruition as most of the bulk eliminated in main modules reappeared in power plant modules producing no significant transportability gains.

- An inherent characteristic of the design (consistent with other GEM vehicles) is that large cargo areas are readily achieved.

11.3 CONCLUSIONS WITH REGARD TO STRUCTURAL DESIGN

- Main structural members in general use are built-up beams, thin-webbed, disposed both longitudinally and transversely, and typified by the techniques of aeronautical structural practice.

- No novel approach has been uncovered in any structural activity which offers any prospect for improved structural efficiency; neither has this program yielded any unique solution or state-of-the-art advance.

- Measurement of structural efficiency is complicated by the variation in criteria and loads used by different design teams and by the diversity of secondary structure appearing in the various designs.

- Structural weights are found to spread from 20-40 per cent of gross weight -- the heavier structures in general being designed to more rigorous load conditions.

- The design as evolved in this study has a structure-gross weight relation of 40 per cent but since it has minimum secondary structure, and simplified facilities throughout and the modular weight penalty makes a small contribution to the figure, the fact that the percentage is on the high end of the scale must be attributed to the realistic structural criteria proposed.

- Maximum environmental resistance in the amphibious regime is not compatible with a structure of minimum weight; the preferred structural material on an overall basis has a specific strength of only about 60 per cent of other materials that might be employed.

- In proposing structural criteria, it is found that the hazards of operation in the military environment override the cases which are derived from the motion of the vehicle in normal operation.

11.4 RECOMMENDATIONS

With the one qualification as noted below, BAARINC considers that the objectives of TRECOM in this design investigation have been fulfilled and that there are no technical obstacles to proceeding to the next step in a full scale development of a Ground Effect Machine for the LOTS mission or a related one.

- Accordingly, it is recommended that further design studies in the structural area having a generalized and feasibility aspect are unwarranted and should not be pursued.

- It is recommended further that in a structural development program for a GEM other than one destined to be transported exclusively by ship, that the modular design concept be employed and its advantages utilized.

The question of load cases and load factors and structural criteria in general have not produced any figures or even a philosophy which has general acceptance. As a consequence definitive conclusions on efficiency of structural designs have not been reached. At the present time, operational experiences with second generation GEM's are beginning to accumulate.

- Accordingly, it is recommended that an analysis effort be initiated, which would include actual full-scale operational results under a variety of conditions, to the end that structural criteria -- effective and in useable form -- could be applied to the design of a GEM for the Army's LOTS carrier or the USMC assault and support craft or a vehicle with a related military mission.

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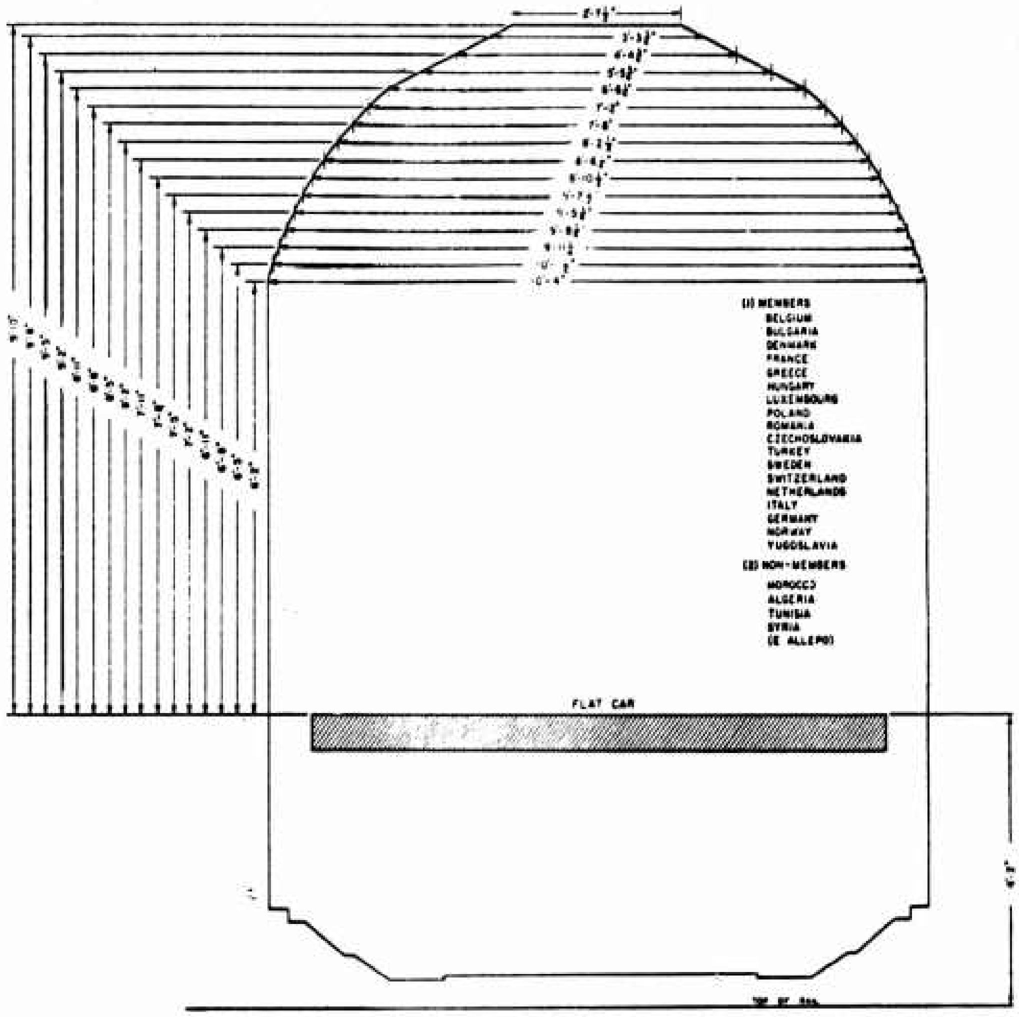
BERNE INTERNATIONAL
(CLEARANCE DIAGRAM)

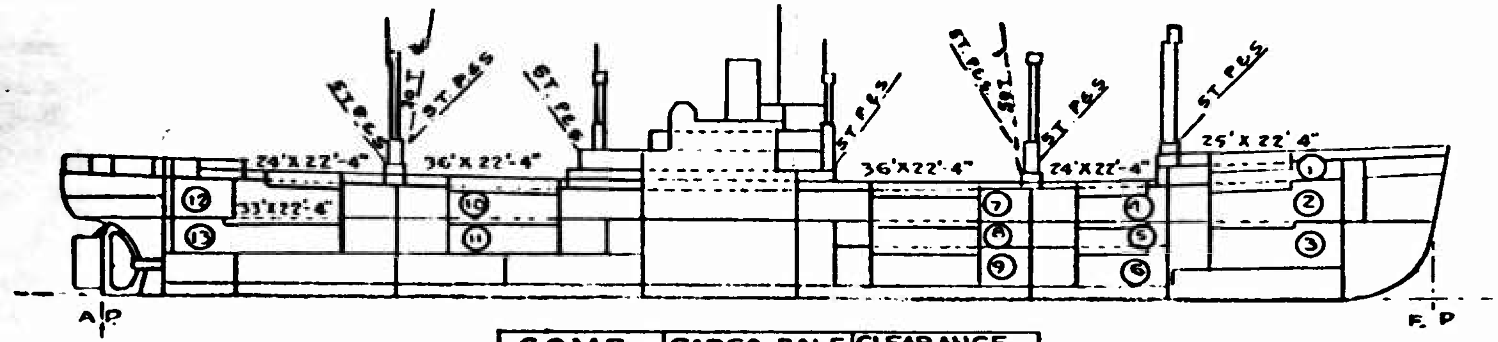


ITEM LENGTH 32' 10"

from AR 705-8

Appendix A

MARITIME ADMINISTRATION DESIGN VC2-S-AP3
VICTORY CLASS



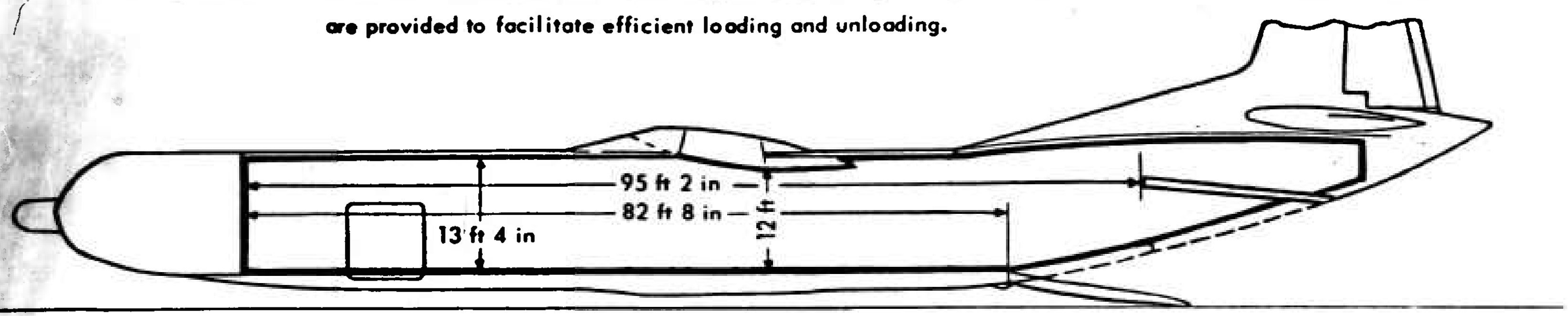
COMP. No	CARGO BALE CUBIC FEET	CLEARANCE UNDER HATCH GIRDERS
1	18 730	6'- 7"
2	23 785	9'- 8"
3	27 910	15'- 4"
TOTAL	70 425	
4	27 010	8'- 5"
5	21 805	7'- 7"
6	27 945	11'- 4"
TOTAL	76 760	
7	45 555	8'- 1"
8	37 795	7'- 7"
9	52 840	11'- 4"
TOTAL	136 190	
10	49 200	8'- 0"
11	51 100	10'- 4"
TOTAL	100 300	
12	43 630	10'- 7"
13	25 905	10'- 4"
TOTAL	69 535	
GRAND TOT.	453 210	

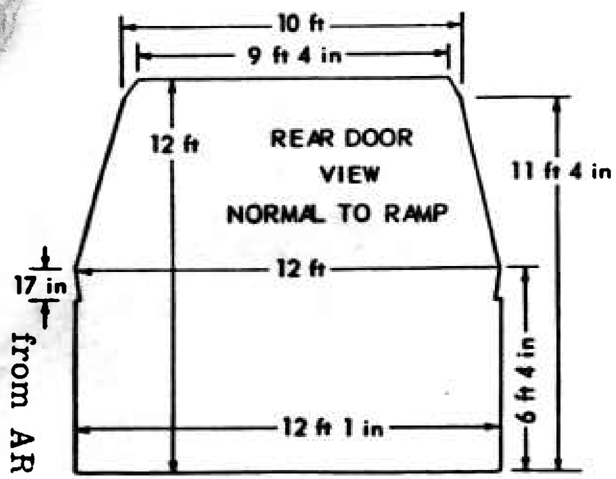
Note:
P - Port Left Side
S - Starboard Right Side
T - Long Tons

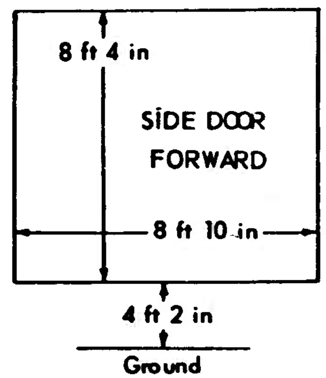
from AR 705-8

C-133 A
CARGO COMPARTMENT PROFILE

DESCRIPTION OF AIRCRAFT : C-133A is a four engine aircraft designed to serve as a long range logistics carrier, transporting cargo primarily. Aft loading door with integral ramp, forward side loading door, and truck-bed height loading are provided to facilitate efficient loading and unloading.







DIMENSIONS:

Main Compartment:

Height (useable)	- 13 ft 4 in
Height (under rear spar)	- 12 ft
Length (overall)	- 97 ft 4 in
Wide, Width (floor level)	- 11 ft 11 in

CAPACITY:

Main Compartment	- 13,028 cu ft
Ramp incline	- 9°
Ramp toe incline	- 15°

from AR 705-8

REFERENCE: (1) Technical Order 1C-133A-9
(2) Standard Aircraft Characteristics Book

TYPICAL LOGISTICAL MISSION : 1,000 Nautical Miles (one way) - Normal weight 95,000 lbs.

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Unclassified Report

Structural configurations, materials and criteria for GEMs in the Army's amphibious regime of transportation

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2. Transportability--Ground Effect Machines.
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